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Survey of Cryogenic Cooling Techniques

Aerospace Corp.

prepared for

Space and Missile Systems Organization

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Survey of Cryogenic Cooling Techniques

Prepared by MARTIN DONABEDIAN
Vehicle Engineering Division

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Engineering Science Operations
THE AEROSPACE CORPORATION

Prepared for SPACE AND MISSILE SYSTEMS ORGANIZATION
AIR FORCE SYSTEMS COMMAND
LOS ANGELES AIR FORCE STATION
Los Angeles, California

THE AEROSPACE CORPORATION

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13. ABSTRACT

A survey was made to determine the state of the art of various methods of cooling to cryogenic temperatures for potential adaptation to spaceborne systems. The study included evaluation of mechanical closed-cycle refrigerators, open-cycle expendable systems, passive radiation concepts which utilize the low temperature of the deep space environment, and thermoelectric devices. The following mechanical refrigeration cycles were evaluated: Vuilleumier, Stirling, Gifford-McMahon/Solvay, Brayton and Claude. Open-cycle expendable systems evaluated were high-pressure gas coupled with Joule-Thomson expansion, cryogenic liquids in both the subcritical and supercritical state, solidified cryogens, and liquids storable at room temperature. Information presented includes background, system characteristics, performance data, and development status and potential. System selection guideline maps are also presented as an aid in determining the most appropriate system for any given application.

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FOREWORD

This report is published by The Aerospace Corporation, El Segundo, California under Air Force Contract No. F04701-72-C-0073. This report, which documents research carried out from 1 November 1970 through 15 May 1972, was submitted on 30 October 1972, for review and approval, to Major Gerald J. Ringes, USAF.

Approved

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D. Willens, Director
Vehicle Design Subdivision
Vehicle Engineering Division

Publication of this report does not constitute Air Force approval of the report's findings or conclusions. It is published only for the exchange and stimulation of ideas.

Gerald J Ringes

GERALD J RINGES, Major, USAF
Deputy Chief, Payload Section,
Surveillance Directorate
Deputy for Technology

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FIGURES (Continued)

I. INTRODUCTION AND SUMMARY

A. INTRODUCTION

This report presents a survey of the state of the art of the equipment and methods used in cooling to cryogenic temperatures for application in spacecraft. This effort was conducted in the Vehicle Engineering Division under Aerospace JOs 6471 and 2441, entitled "Cooling to Cryogenic Temperatures."

Providing refrigeration at cryogenic temperatures for infrared detectors, optical system components, and other devices in spacecraft is an important requirement for many space projects and a challenge for modern technology. A number of potential devices in various stages of development can provide the many cooling requirements; each offers different combinations of weight, power, size, reliability, and in many cases a specific refrigeration capacity range and temperature level for which it is best suited. Because of the wide range of operational requirements, however, in many cases it is difficult to generalize about application of specific cooling devices.

The four fundamental concepts of providing cryogenic cooling to be discussed in this report are:

a. Closed-cycle mechanical refrigerator systems which provide cooling at low temperatures and reject heat at high temperatures

b. Open-cycle expendable systems. This category includes stored high-pressure gas with a Joule-Thomson expansion, the use of stored cryogens either in the subcritical or supercritical liquid state, solid cryogens, and liquids stored at room temperatures.

c. Passive radiators which cool systems to cryogenic temperatures by radiation to the low temperature, deep space environment

d. Thermoelectric coolers utilizing the Peltier cooling effect which is caused by a current passing through a junction of two dissimilar materials

This report is intended as a sourcebook and system selection guideline which may be updated from time to time to include new or revised information. This is especially reflected by the section on closed-cycle mechanical refrigerators in which the format for tables and charts is arranged to facilitate continual addition of new data. Although brief descriptions are given of the thermodynamic cycles associated with many of the closed-cycle refrigerators, no attempt has been made to describe the cycles in detail or to derive the ideal thermodynamic efficiency of any cycle. For this type of theoretical information, it is recommended that the reader obtain more basic references, such as those listed in the bibliography of this document.

B. SUMMARY

Background data, description and operation of various refrigeration cycles, performance data, and development potential have been summarized for a number of refrigeration concepts adaptable to spaceborne operations. Because of the extensive data compiled in this document, the highlights of the entire report are described in this summary section. Various concepts of providing cooling at cryogenic temperatures are briefly defined, some of the more significant operating characteristics are indicated, and basic integration limitations with the entire spacecraft are outlined. Since there are so many variables and design tradeoffs involved when considering a system for a specific application, it is not considered practical or even meaningful to attempt to define the exact operational regimes where specific systems are optimum. However, there are general operational areas (e.g., temperature and cooling load) for which specific systems can be determined to be most appropriate and for which general guidelines can be provided for feasibility studies and preliminary design purposes.

1. SYSTEM DEFINITIONS AND OPERATING CHARACTERISTICS

The various refrigeration systems covered in this report have been categorized into four fundamental types: closed-cycle mechanical refrigerators, open-cycle expendable systems, passive radiative coolers, and thermoelectric coolers.

The simplest and potentially most reliable method of providing cryogenic cooling is to utilize the low temperature sink of space directly by using a radiator. The concept is attractive since the system is passive, requires little or no power and is capable of high reliability for extended periods. The radiator must be shielded against direct sunlight and the parent spacecraft and, in the case of near-earth orbits, heat inputs from direct thermal emission and reflected sunlight from the earth and its atmosphere. Primary limitations of this approach are the rapid increase in radiator size with decreasing temperature and the parasitic heat leak into the radiator.

Thermoelectric coolers are appropriate to provide cryogenic temperatures for small wattage heat sources. Based on the Peltier cooling effect arising from passing a current through a junction of different materials, thermoelectric coolers provide a simple, lightweight, reliable method of cooling. These systems are limited primarily by the low efficiencies (in the order of one percent) and maximum operating temperature difference between the hot and cold junctions.

For operation at lower temperatures and/or higher cooling capacities, open-cycle expendable systems are appropriate. The simplest, least expensive approach is to use a Joule-Thomson (J-T) cooler wherein a high pressure gas (in the range of 1000 to 6000 psia) combined with a J-T expansion valve (consisting of an orifice, small diameter tube heat exchanger, shield, etc.) results in cooling of the gas and ultimately provides a source of liquid at the point to be cooled. The utilization of helium, hydrogen, argon or nitrogen enables developed units to provide cooling from approximately 4.2 to 77 K at capacities in the range of 0.50 to 10 W. The primary limitation is the high weight penalty for storage of high-pressure gas (approximately 4 lb per pound of gas for N_2 and 15 for H_2). One advantage of this approach over cryogenic storage is the ability to provide intermittent operation over a long period of time.

Cryogenic fluids stored as liquids in equilibrium with their vapors (subcritical storage) can provide a convenient constant temperature control system for ground-based or advanced aircraft and spacecraft applications. A variety of fluids are available which provide temperatures ranging from 4.2 (helium) to 240 K (ammonia). The primary limitation of this approach is the complex tank design required to minimize boiloff and the direct relation of weight and volume requirements with elapsed time.

The same fluids can be stored at pressures above their critical pressures (supercritical storage) as homogeneous fluids, thus eliminating phase separation problems encountered during weightless conditions in space. A weight penalty as compared with subcritical storage normally accrues as a result of the higher operating pressures required. The added flexibility of a single phase homogeneous fluid makes this approach competitive, in some cases, with closed-cycle mechanical refrigerators for missions of 60 to 90 days.

The sublimation of a solidified cryogen can provide reliable refrigeration for small wattage heat sources for periods measured in months to a year or longer. This approach utilizes a solidified cryogen in conjunction with an insulated container, an evaporation path to space, and a conduction path from the coolant to the device being cooled. Advantages over the use of cryogenic liquids include a higher heat content per pound of coolant, and a higher density storage resulting in less storage volume, and the lower temperature solid phase can permit a gain in sensitivity in certain infrared detectors. Temperatures ranging from 10 K (using hydrogen) to 90 K (using methane) and 125 K (using CO_2) or higher are achievable. Limitations involve restrictions on detector mounting, specialized filling procedures, and temperature control requirements.

A limited number of fluids exist which can be stored at ambient temperature (thus eliminating the boiloff problem of cryogenic fluids) and thermodynamically manipulated to provide cooling at about 100 K. However, development is required to advance this concept to the hardware stage.

To provide cooling in the range of approximately 4 to 100 K or higher in capacities ranging from a fraction of a watt to 100 W for periods of months to years, closed-cycle mechanical refrigerator systems may be required. The most significant components of mechanical refrigerator systems involve a power supply, power conditioning equipment, the refrigerator itself, and the heat rejection system which normally includes a remote radiator and a heat transport loop. A number of different refrigerator cycles are in various stages of development for space applications, but none currently have demonstrated the capability to operate maintenance-free in excess of 2000 hr. The high penalty for spaceborne power places a premium on refrigerator efficiency. Reliability also is of primary importance.

Stirling and Vuilleumier (VM) cycle systems possess several of the primary requirements for spaceborne refrigeration systems. The primary area of application of these systems is at temperatures above about 10 K and for capacities below 50 to 100 W. Both of these cycles can be classed as intermittent flow systems and utilize thermal regenerators to store and release energy during the completion of each cycle. As the specific heat of all materials becomes very small near absolute zero, the effectiveness of storing thermal energy, and thus the efficiency of both these cycles, becomes very poor at temperatures below about 10 K.

Gas-bearing supported turbomachinery utilizing reversed Brayton and Claude cycles is being developed for use with very low temperatures and/or high capacity systems. Turbomachinery units appear to have the best potential for long life. However, the high power requirements (due primarily to low efficiencies of turbocompressors and expanders) make these systems competitive only at temperatures below about 20 K or at higher capacities where the component efficiencies improve.

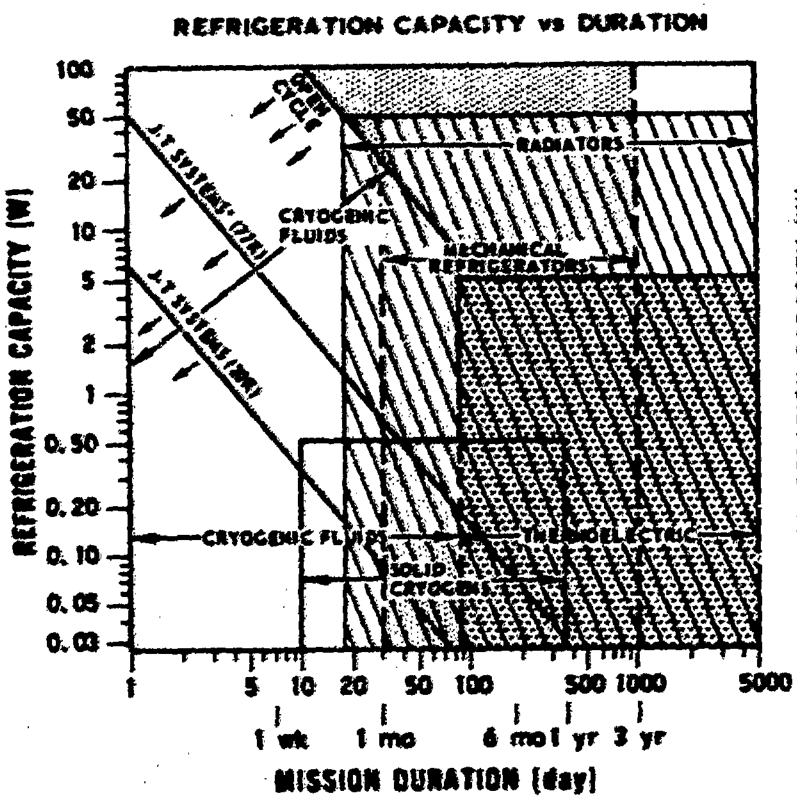
Rotary-reciprocating machinery utilizing the Brayton or Claude cycle (in which portions are rotated as well as reciprocated) also shows promise for long life and potentially provides a minimum power system at temperatures

below about 20 K. Both turbomachinery and rotary-reciprocating units are being developed under Air Force contracts.

2. CRYOGENIC SYSTEMS SELECTION GUIDELINES

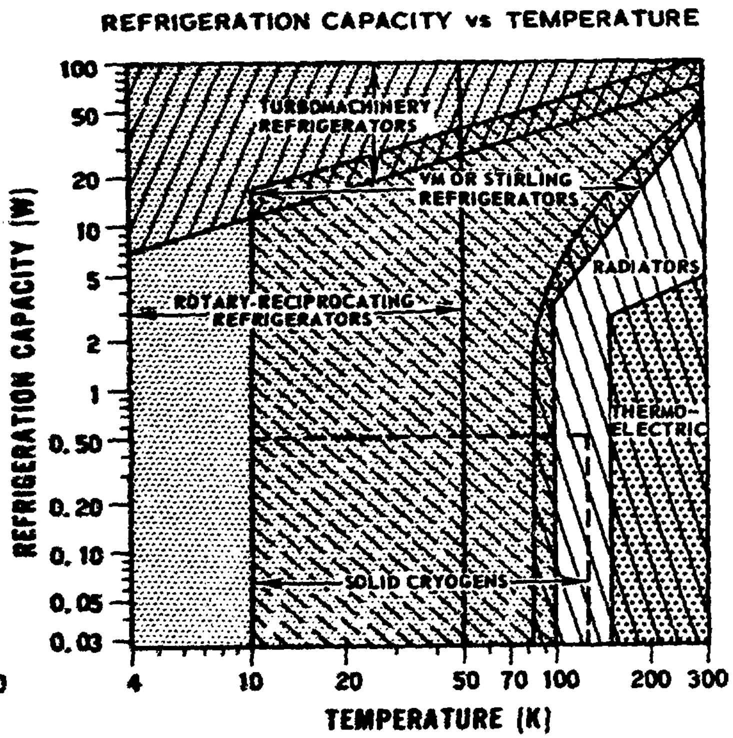
To aid in selecting the type of cryogenic refrigeration system for use in a given application, two system selection maps have been prepared (Figs. 1-1 and 1-2). Three primary variables (temperature, refrigeration capacity, and mission duration) are required in most cases to properly identify the most desirable system. Figure 1-1 shows refrigeration capacity in watts versus mission duration in days, while Fig. 1-2 shows refrigeration capacity versus refrigeration temperature (K). It should be clearly noted that these merely represent guidelines and in many cases, especially where regimes are in close proximity or even overlap, additional criteria such as booster payload capability, geometry limitations, type of orbit, the reliability required, the development cost and time available, etc., will determine which system is most appropriate. These charts are based on the technology that either exists or is under development to the point at which a given system can be applied in the next two to three years. Figure 1-1 establishes the fundamental concepts that can be considered and Fig. 1-2, along with various charts in the report, defines the specific system.

The upper bounds shown in Fig. 1-1 for open-cycle systems are based on weight limitations associated with a 3000- to 4000-lb spacecraft and a cryogenic storage limitation of 90 days. However, tankage currently under development indicate durations of six months to a year and longer may be feasible. Open-cycle systems utilizing cryogenic fluids are applicable at essentially any point within the capacity and temperature boundaries of Fig. 1-2 and are limited primarily only by weight (as determined by mission duration and refrigeration capacity). Specific fluids available are shown in Table 3-3. For very short durations and capacities of 10 to 20 W or less, high-pressure gas systems coupled with J-T expansion valves provide the simplest, most economical means of refrigeration. However, the weight penalties for high-pressure gas storage limit the practical duration. Solid



*High pressure gas storage with J-T expansion

Fig. 1-1. System Selection - Refrigera'ion Capacity Versus Duration



Note: Open-cycle fluid systems are applicable at any point on this chart because they are limited only by duration

Fig. 1-2. System Selection - Refrigeration Capacity Versus Temperature

cryogenic systems are most attractive for capacities of less than about 0.50 W for temperatures ranging from about 10 K (hydrogen) to about 125 K (CO_2) but are feasible for substantially larger loads. Characteristics of various solid coolants are summarized in Table 3-9. Durations of one month up to a year or longer appear to be feasible.

For temperatures above about 100 K passive systems utilizing radiators become attractive, especially where long mission durations are involved; this may preclude the use of mechanical refrigerators. As the temperature increases, the radiator becomes more effective due to the fourth power radiation factor and is competitive at larger refrigeration capacities. Systems in various degrees of development have been designed for operation in the range of 80 K (5 to 10 mW) to 170 K (10 W). Beyond about 150 K, and for loads less than about 2 to 5 W, thermoelectric systems become competitive, although the most attractive area of application is in the fractional wattage load range. Nearly all thermoelectric systems designed in the past have been based on a 300 K sink temperature with cold junction temperatures down to about 200 K. Prototype systems developed for military application operate at a minimum temperature of 170 K while experimental systems under development have been operative as low as 145 K under no-load conditions. If these devices are coupled to radiators operating at 200 K, for example, and with materials currently under development, thermoelectric systems operating in the 125 K region may be feasible. Characteristics of typical thermoelectric coolers are presented in Table 5-2.

Mechanical refrigerators are applicable over a wide range of conditions from about 4 K to near 100 K at moderate capacities and possibly to higher temperatures at relatively higher capacities. Many specific applications may exist at capacities from a fraction of a watt up to 100 W for mission durations ranging from a month to possibly three years and beyond as limited by refrigerator capabilities.

Stirling cycle systems developed and operational in aircraft (and one spacecraft) are the most efficient in terms of power requirements down to about 10 K (as limited by regenerator efficiencies) at moderate capacities.

Maintenance-free operational life of 1000 hr has been demonstrated. A slightly modified commercial unit has been utilized in space. Vuilleumier cycle systems recently developed have inherent capability for longer life than Stirling cycle units and have the potential advantage of being powered directly by a heat source. A VM unit was recently successfully operated in space, while a number of units are in various stages of development for aircraft and spacecraft uses.

Turbomachinery or rotary-reciprocating machinery utilizing Brayton or Claude cycles are applicable from temperatures of around 4 K up to 40 or 50 K. The lower efficiencies associated with the turbomachinery systems will generally restrict applications to the larger capacities. Both of these systems may have potential for extremely long life. Components have been developed and fabricated; however, no significant overall system performance data or operating experience are available.

Gifford-McMahon/Solvay systems which utilize separable components have been fully developed for aircraft systems and currently possess the longest maintenance-free operating life of about 3000 hr. Low efficiencies, however, result in significantly more power and weight than comparable VM or Stirling systems and thus these systems appear to have less promise for space applications.

Characteristics of 75 refrigerators are summarized in Table 2-14.

II. CLOSED-CYCLE MECHANICAL REFRIGERATOR SYSTEMS

A. BACKGROUND

This section includes the description, operating characteristics, and state of development of various types of mechanical refrigerators applicable to spaceflight use. With few exceptions, these existing refrigerators are not immediately suitable for spaceflight, and a number of research and development programs have been pursued by various U.S. Government agencies in order to reduce or eliminate the basic shortcomings of the various refrigerators. Some of these programs have been general in nature and have been intended to raise the overall level of technology while others have aimed at developing a refrigerator for a particular mission. The basic constraints or considerations which differentiate spaceborne refrigerators from industrial or airborne systems are the extreme need for low weight and power, and the requirement for extended reliability without opportunity for maintenance and/or repair. The total weight of these systems includes not only the refrigerator itself but the weight of power supply, control equipment and the heat rejection system. Although the specific weights of these auxiliary systems are not discussed here, added emphasis is placed on power requirements in this report because the weight penalty for power can be extremely significant for spaceborne systems.

B. SECTION FORMAT

The primary objective of this section is to summarize current types of refrigerators that can be applied to space operations. In order to facilitate future revisions, the mechanical refrigerator section has been broken down into five basic categories: Vuilleumier, Stirling, Gifford-McMahon/Solvay, Reversed Brayton/Claude (further separated into turbomachinery and rotary-reciprocating systems) and Joule-Thomson closed-cycle systems. (Open-cycle J-T systems are covered in Section III-A.) Each of these categories

includes basic descriptions and operating characteristics, data relating to current manufacturing and development efforts, performance data, specific weight and power per watt of refrigeration, and potential development problems. In each category a brief listing of existing refrigerators (including development, prototype and production units) is provided. At the end of the section on closed mechanical refrigerators, a master list (see Table 2-14) of all refrigerator units with additional system characteristics is also provided. The technique of assigning a specific identification number to each refrigerator unit, even though it may be only a development or prototype system, was utilized to differentiate between similar systems and to facilitate addition of new data.

C. VUILLEUMIER (VM) CYCLE REFRIGERATOR

1. BACKGROUND AND DESCRIPTION

The Vuilleumier (VM) thermodynamic cycle is a heat-driven cycle patented by Rudolph Vuilleumier in 1918. This constant volume cycle operates through the use of displacers which have the advantage of requiring minimum seals as the pressures throughout the system are nearly equal at any moment. The displacers simply move the gas from one section to another without the requirements of compressing gas within a closed volume. This results in minimal loading on bearings and seals with inherent long life potential for such a system.

A schematic of a typical VM reciprocating refrigerator is shown in Fig. 2-1a. The refrigerator consists of two sets of cylinders with displacers arranged to cycle approximately 90 deg out of phase and a small electric motor to drive the displacers. A thermal regenerator in each displacer (or external to it) allows the gas to move from one end of the cylinder to the other while alternately storing and releasing thermal energy. Heat is supplied at the hot end of the power cylinder and rejected at the ambient (crankcase) end. Heat from the cryogenic load is absorbed at the cold end of the refrigerator cylinder and also rejected at the crankcase. A cross-section of a typical single stage VM prototype refrigerator is shown in Fig. 2-2.

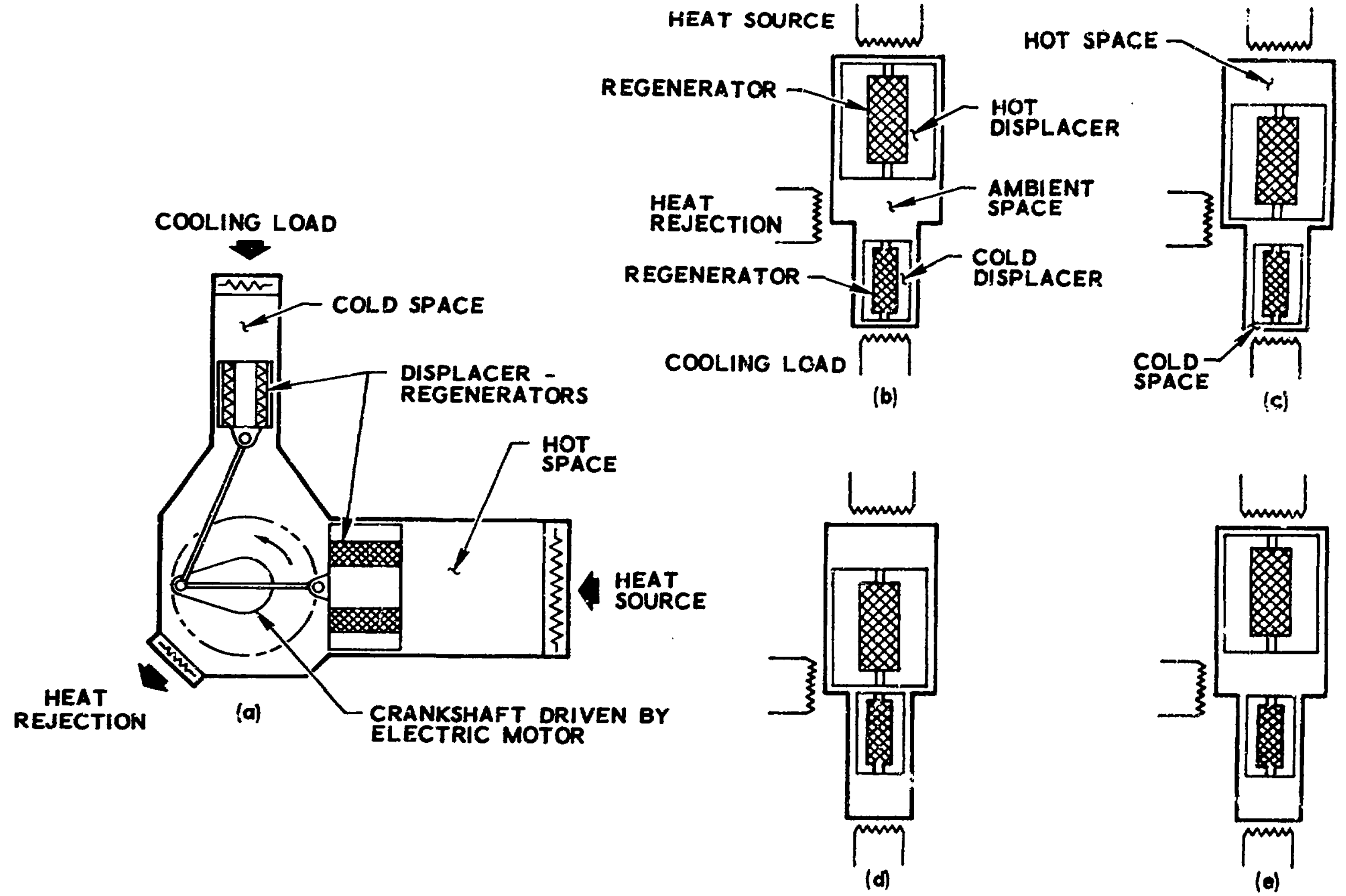


Fig. 2-1. VM Cycle Refrigerator Concept

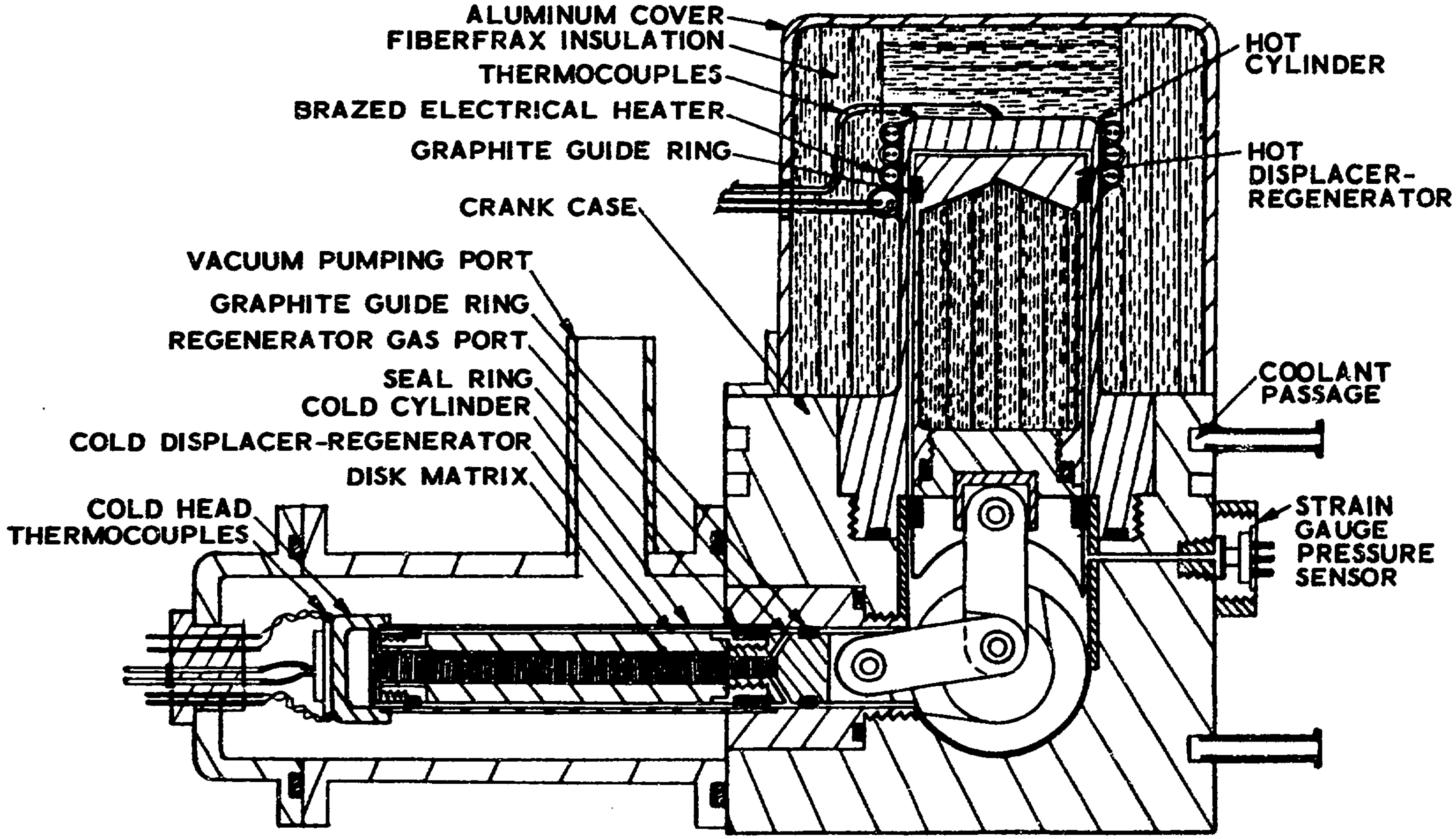


Fig. 2-2. VM Cycle Prototype Refrigerator Cross-Section (Ref. 1)

The operation of the VM cycle can best be visualized by considering a simplified schematic showing four specific positions of the two displacers as shown in Figs. 2-1b through 2-1e. In position b the fluid is primarily in the ambient space. From b to c the hot displacer moves from the hot end to the ambient end, causing fluid to move from ambient to hot spacers at constant volume, resulting in an increase in system pressure and hence compression of the fluid remaining in the ambient space. From c to d the cold displacer is moved to displace this remaining fluid to the cold end. From d to e the pressure is reduced by displacing fluid from the hot space back to the ambient space, thereby expanding the fluid in the cold space. From e to b the cold gas is returned to the ambient space by movement of the cold displacer to begin a new cycle. In practice the intermittent movement of the displacers is achieved by driving both of them from the same crankshaft as in Fig. 2-1a but displaced in phase such that during compression most of the fluid is in the ambient space, and during expansion most of the fluid is in the cold space.

Two of the primary advantages of the VM cycle are the absence of a mechanical compressor and the capability of utilizing direct thermal energy as the primary source of energy input. As a result, a radioisotope heat source, for example, can be used directly without the need for supplying or generating any significant amount of electrical energy. The VM refrigerator is very suitable for closed-cycle cryogenic refrigeration in small capacities applicable to both airborne and spaceborne systems.

The VM refrigerator potentially is a very compact, high-performance refrigerator that can produce refrigeration at cryogenic temperatures for long periods of time without maintenance. It can be powered by direct solar energy, electrical heating, exothermic chemical reactions, a gas burner, or by nuclear energy. The refrigerator can be designed for ground, space or airborne operation. Its size and weight, however, may render it impractical with respect to other cryogenic cooling systems when the required refrigeration power is above about 100 W. Some advantages and disadvantages of VM refrigerators are listed below:

a. The thermal efficiency and the coefficient of performance of the VM cryogenic refrigerator are relatively high because of the thermodynamic cycle employed and low mechanical friction. These are important features in any application.

b. The refrigerator can be designed so that it utilizes thermal energy to produce both refrigeration and a net power output if desired, which can be used for operating other equipment. Any practical heat source is acceptable that can supply the desired temperature and thermal energy at the power cylinder. Thus, this design can provide a compact power-and-refrigeration producing unit with no requirement for an external source of electrical power.

c. The VM refrigerator has a low wear rate on seals because the pressure difference across a dynamic seal in the machine is small. This is particularly true of a slow running unit that is designed so that no net power output is produced. This type of unit is driven with a small electrical motor in order to overcome the internal friction, and the low wear rate gives long maintenance-free operation.

d. The refrigerator is an excellent spot cooler at relatively low refrigeration power levels because it is possible to obtain good performance even in very small size units. However, cooling can also be supplied to remote points by using an external cooling loop which connects the cold cylinder of the refrigerator and the object to be cooled by means of a circulating coolant.

e. The practical VM refrigerator is capable of producing refrigeration down to about 10 K. At lower temperatures the size of the thermal regenerators becomes very large, since the value of the specific heat for all materials approaches zero when the absolute temperature approaches absolute zero. This is a disadvantage to all refrigerators using regenerators. The regenerators must have a high effectiveness or the performance of the refrigerator will be poor.

f. The performance of the VM cycle is strongly influenced by the total dead volume in the refrigerator since it causes the thermodynamic expansion-and-compression paths of the gas in the refrigerating cylinder to come closer to each other. Thus, the result is a thin pressure-volume diagram and a smaller cooling power when the dead volume is increased. The effect of an increase in the dead volume is very pronounced in smaller refrigerators, and every effort must be made to keep dead volume as low as possible.

2. DEVELOPMENT STATUS OF VM REFRIGERATORS

Five manufacturers are currently involved in the development of VM refrigerators: Philips Laboratories, Garrett-AiResearch, Kinergetics (formerly the Submarine Systems Division of Sterling Electronics), the Hughes Aircraft Company, and RCA.

Philips has recently built and successfully tested two small prototype units under contract to the Army Night Vision Laboratory at Ft. Belvoir, Virginia. The units have been built with the application of military infrared systems in mind, utilizing their quiet operation to advantage. Philips has delivered one of these units to the Army for testing. This 10.3-lb cooler which provides 0.50 W of cooling at 77 K for approximately 90 W of electrical power input was designed for 1000 hr of maintenance-free operation in a terrestrial environment with a projected MTBF of 3000 hr. During initial tests one failure occurred at 450 hr due to faulty regenerator pads while a second failure occurred at 1150 hr due to faulty heater brazing. The Army is ultimately interested in using a chemical heat source in place of the electrically-powered hot cylinder. Philips is also under contract to the Air Force Flight Dynamics Laboratory (AFFDL) to develop and build two VM units for space application. These units are to provide simultaneous cooling at three stages ranging from 11.5 to 75 K for up to 5000 hr.

Garrett-AiResearch presently is under contract to NASA, Goddard Space Flight Center, Greenbelt, Maryland, to develop and test a VM refrigerator to deliver 5 W of cooling at approximately 75 K with a lifetime goal of two to five years. This development is part of the Integrated Isotope-Cooling Engine System (ICICLE) program. Initially, the purpose of this program was to develop a radioisotope*-driven VM refrigerator for use in meteorological and communications satellites; however, all development and testing will be conducted with electrically powered energy sources. AiResearch has fabricated a number of laboratory models and delivered an engineering

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*Pu-238

model to NASA for testing in late 1971. The design requirement in terms of power input is 350 W, whereas there is no weight requirement. Based on Ref. 2, a model that was delivered to NASA will produce about 7 W at 75 K (to provide a margin for expected degradation) with a 350-W input.

Kinergetics has been in the VM development area about two years. Work on VM units included programs under contract to AFFDL for development of a space flight unit (0.50 W at 5 K), which was subsequently terminated, and to the Army (Ft. Belvoir) in the area of night vision (0.40 W at 77 K), contracts F33615-70-C-1130 and DAAK 02-70-C-0436, respectively. The Army unit has been delivered for testing.

Hughes Aircraft has built a number of experimental and prototype units for ground, aircraft and spacecraft applications under contract to AFFDL and to the Army. A few of these early experimental models are no longer active; however, a number of others are in various stages of development and/or testing. Three of these units were developed for aircraft application. One unit, designed for an advanced forward-looking IR system (ID #22), provides simultaneous cooling at 25 and 75 K and has recently completed flight testing. Another unit designed for an aircraft IR scanner system (ID #26) provides cooling at 77 K and has accumulated over 1100 hr of laboratory testing at AFFDL. A third unit (ID #25), developed for a missile guidance application which provides cooling at 85 K, has been delivered to the AFFDL and has undergone brief testing.

A 15 K spaceborne refrigerator was developed under AFFDL contract (completed) for a celestial measurement sensor. This unit was successfully flown on Air Force Flight SESP 71-2 and accumulated approximately 1000 hr of operation including about 450 hr on-orbit before the unit was shut down due to a supporting subsystem malfunction. Hughes is currently under contract to AFFDL (in competition with Philips) to develop and build two units which provide three stages of cooling from 11.5 to 75 K to operate for a period of 5000 hr. Hughes is also involved in the Army Night Vision Laboratory program to develop a single-stage VM refrigerator to provide 1000 hr of

maintenance-free operation in a terrestrial environment. One such unit (ID #40), weighing 6.5 lb and providing approximately 0.60 W at 77 K with 105 W of input power, was delivered late in 1969 to the Night Vision Laboratory for testing.

The RCA Defense Electronics Laboratory (Camden, New Jersey) has developed two VM coolers. The initial effort of RCA in this area was an in-house development of a laboratory model VM (2.5 W at 77 K) intended for the NASA ICICLE program. This unit has not proceeded past the experimental stage. A second VM cooler is currently under development for the Army Night Vision Laboratory as a portable field unit. This is a thermally-driven unit using a propane heater and has accumulated approximately 120 hr of testing. The unit weighs less than 20 lb, provides 1.6 W at 77 K and is scheduled for delivery to the Army sometime in the middle of 1972 for further testing.

3. PERFORMANCE DATA

Because of the development status of VM refrigerators, relatively little performance data are available. Data on prototype units from the various manufacturers are summarized in Table 2-1. In some cases in which sufficient test data could not be obtained, design goals or projected performance estimates were utilized. In most cases, sufficient information was available to establish power input requirements; however, weight data were harder to obtain. Specific weight and power data available are plotted in Fig. 2-3. More than one data point for a given refrigerator design indicates a multistage system; power and weight ratios have been calculated for each stage. It should also be noted that system weight represents only the refrigerator system and does not include the equivalent power or heat rejection system weights.

4. POTENTIAL DEVELOPMENT PROBLEM AREAS

The basic problem areas associated with the development of VM refrigerators have been previously evaluated and summarized by M. Bello (Ref. 3).

Table 2-1. Identification of Vuilleumier Cycle Refrigerators

Identification Number	Manufacturer or Developer	Model, Description, or Program	Operating Temperature Range (K)	Typical Refrigeration Performance	Status	Power: Input (W)	Power: Input/Refrig. (W/W)	Weight: Total (lb)	Weight: Tot/Refrig (lb/W)	Remarks	Ref.
21	Hughes	Development	15 - 75	0.15 W at 15 K	Prototype no longer exists	170	2460	----	------	Early experimental model	4
22	Hughes	AFFDL Development (AFLIR)**	25 - 75	2.0 W at 25 K 1.0 W at 75 K	Flight test completed	1200	600 400	19.5	9.75		5, 6
23	Hughes	Development	10 - 75	0.50 W at 10 K	Prototype no longer exists	550	1100		------	Early experimental model	6
25	Hughes	AFFDL Development for Missile Guidance	70 - 95	0.20 W at 85 K	Two units delivered to FDL for lab. testing	180	900				6, 7
28	Hughes	AFFDL Development for Aircraft IR Scanner	~77	1.5 W at 75 K	Accumulated 1100 hr testing at FDL	200	133	5.75	3.84		6, 8
39	Philips	Development	77 - 280	0.5 W at 77 K	Prototype	70	140	8	16.0		9
40	Hughes	Army Night Vision Lab Development	50 - 110	0.6 W at 77 K	Prototype tested	105	175	6.5	10.8		10
43	Hughes	X 442950 (Early model for ID #42)	Two Stage 10, 75	0.50 W at 10 K 6.0 W at 75 K	Inactive	480	960 80	9.6	19.2 1.6	Simultaneous loading at two stages	6, 11
42	Hughes	SAMSO Experiment 602 for SESP-712	Two Stage 15, 60	0.15 W at 15 K 3.5 W at 55 K	Experimental Flight Model (completed one flight)	540	3600 154	60	400 17.4	Simultaneous loading at two stages	12
41	AiResearch	Development, ICICLE Program	~75	5.0 W at 75 K	Prototype	350	70			Power based on design goal	1
44	Kinergetics*	AFFDL Development (cancelled)	~5	0.5 W at 5 K (Design Goal)	Terminated (uncompleted)	1000	2000	----	------	Only components made	6
51	Philips		~77	1.0 W at 77 K	Prototype	120	120	15	15.0		13
62	Kinergetics*	Army Night Vision Lab Development	77	0.40 W at 77 K	Prototype Delivered	65	163	6	15.0		14
69	Hughes/Philips	AFFDL Development	11.5 - 75	0.30 W at 11.5 K 10.0 W at 33 K 12.0 W at 75 K	Under Development	2700	9000 270 225	130	435 13.0 10.8	Simultaneous loads at all three stages	--
71	Philips	Army Night Vision Lab Development	60 - 100	0.5 W at 77 K	Completed 1000 hr testing	90	180	10.3	20.6		10
72	RCA	Army Night Vision Lab Development	77 K	1.6 W at 77 K	Prototype delivered to Army in Feb 1972	----	----	16	10	Thermally driven with a propane heater	15

*Formerly Submarine Systems

**Advanced Forward Looking IR System

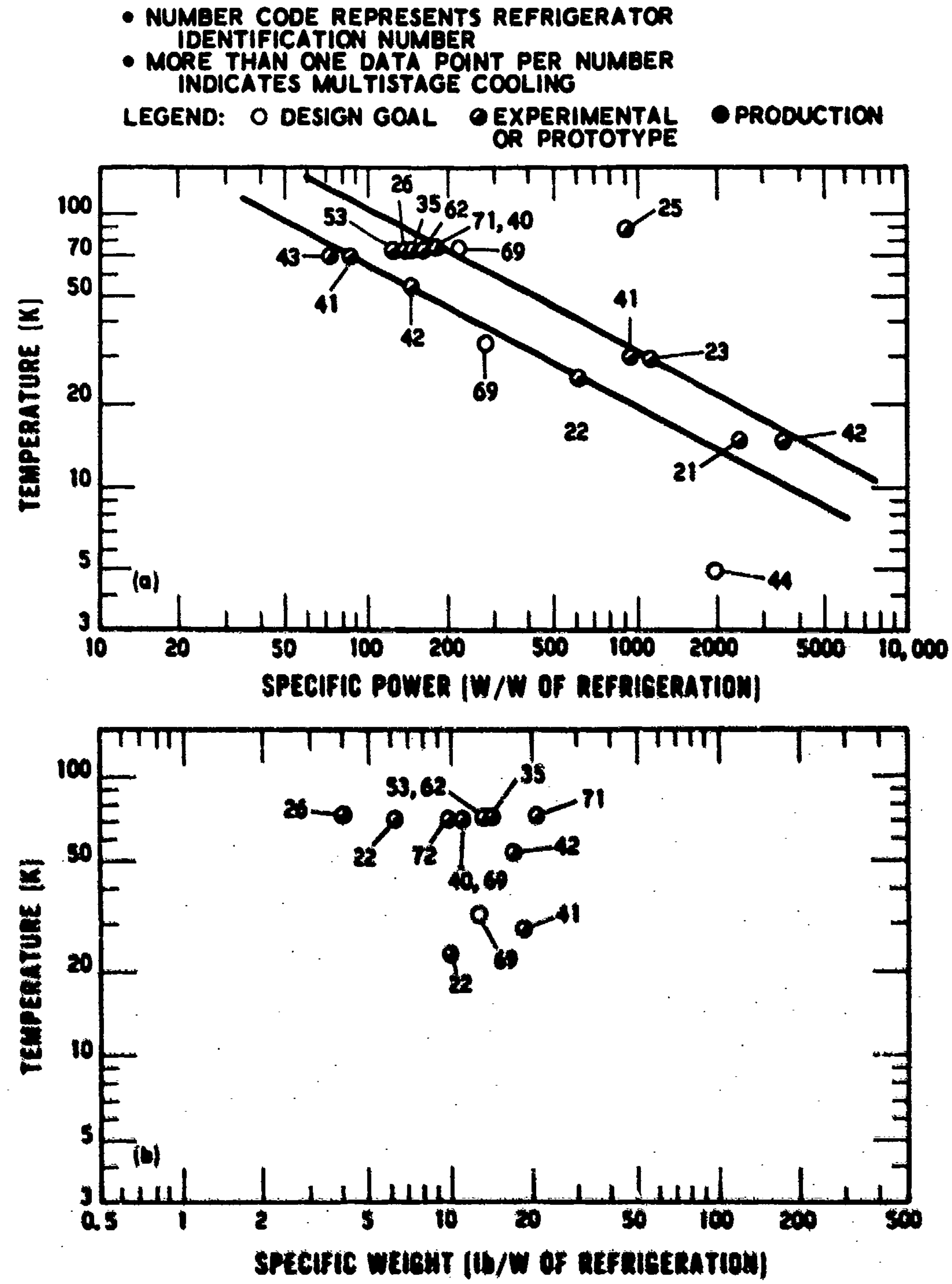


Fig. 2-3. VM Cycle Refrigerator Power and Weight Characteristics

This information is summarized in Table 2-2. The principal degradation mode of the VM refrigerator was identified in the above study as the failure to maintain the required temperature which is traced to two causes: (1) contamination of the refrigerant and (2) seal wear. Contaminants gradually accumulate in the refrigerant and freeze out in the regenerator thereby reducing regenerator effectiveness. It is concluded from the above study, however, that there are no life-limiting component problems which cannot be solved by development.

D. STIRLING CYCLE SYSTEMS

1. BACKGROUND AND DESCRIPTION

Stirling cycle refrigerators possess several of the primary requirements of a spaceborne refrigerator system such as low power consumption and small size and weight. This cycle employs two isothermal and two constant volume processes as shown in Fig. 2-4. The Stirling system contains no valves, and compression and rejection of heat takes place in one cylinder with a heat exchanger while the heat absorption takes place in another cylinder. A practical Stirling cycle refrigerator is schematically shown in Fig. 2-5.

The operation of the refrigerator shown in Fig. 2-5 is described below. In position 1, the working fluid occupies the ambient space, after-cooler and regenerator. From 1 to 2 the fluid is compressed by inward motion of the compression piston. From 2 to 3 the compressed fluid is transferred from the ambient end to the cold end at constant overall volume by equal incremental displacement of both pistons. During this transfer heat of compression is rejected to the after-cooler, and the temperature is reduced to the cold end temperature in the regenerator. With the fluid now occupying the cold space, cooling load heat exchanger, and regenerator, the fluid is expanded by outward movement of the expander piston, 3 to 4. The fluid is returned from the cold end to the ambient end at constant volume by equal

Table 2-2. Vuilleumier Refrigerator Problem Areas (Ref. 3)

Components	Problem Areas	State of Development	Number of Components
1. Regenerator	Degradation characteristics due to aging and contamination require further assessment-organic spacer material	2	4
2. Counterflow Heat Exchanger	Manufacturing tolerances may shift flow characteristics	1	1
3. Heat Coils	Assembly defects and aging may cause performance anomalies	3	2
4. Displacer Seals*	Wear of displacer seals causes both contamination and leakage	4	2
5. Crankshaft Seal*	Use of rolling seals will require development of working fluid resupply due to leakage through the polymeric seal material	4	2
6. Rotary Motor	Needs life test to verify that wear is not a problem	1	1
7. Pistons (Displacers)	Large temperature gradient along the length of pistons requires special material	2	2
8. Drive Bearings	Requires oil lubrication for extensive life	2	6

*Currently life limiting, correctable by development.

Status Key:
- 0 Fully developed
- 1 Requires life test
- 2 Performance testing
- 3 Currently being developed
- 4 Requires development
- 5 Beyond state of the art (1975)

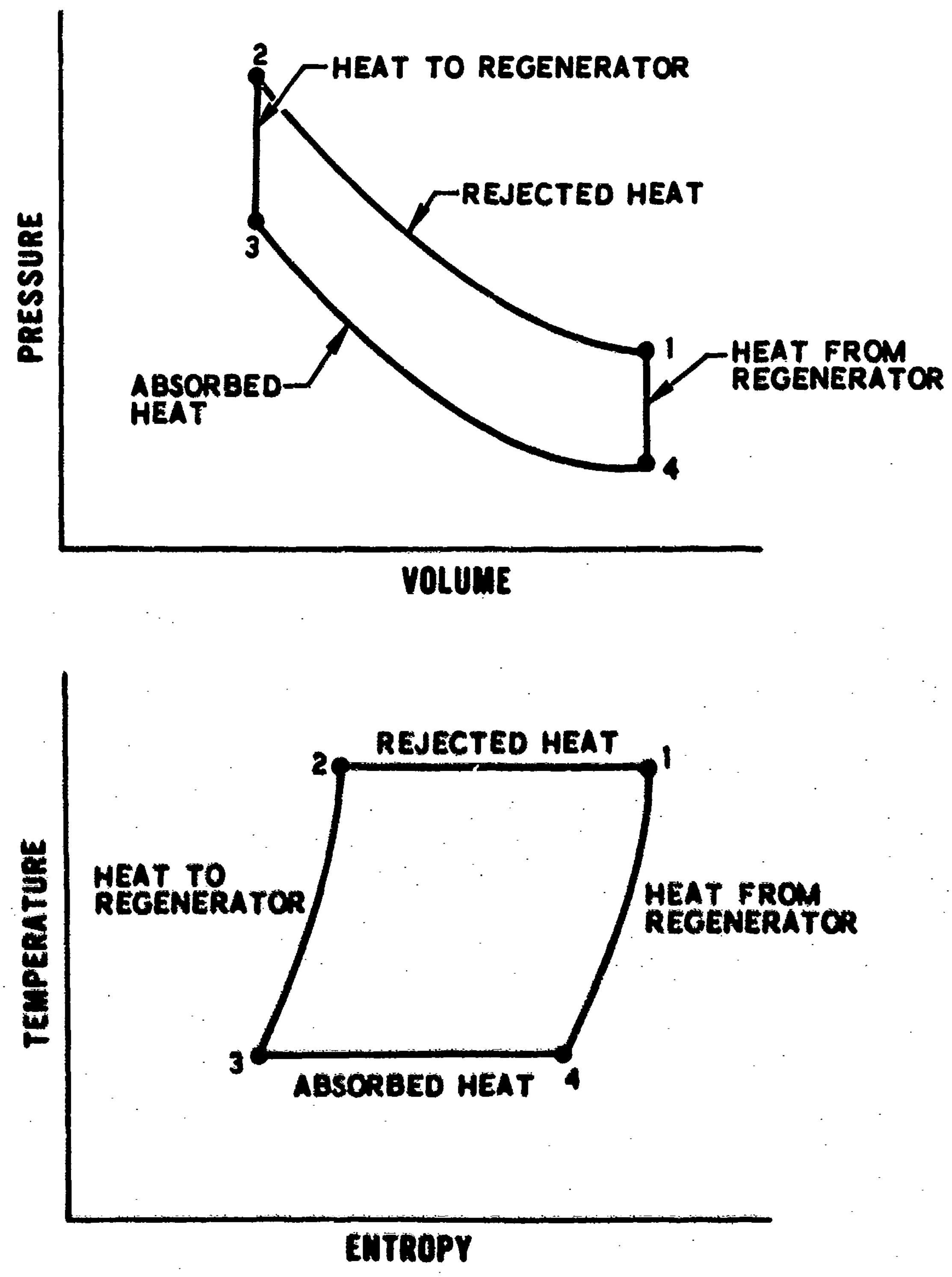


Fig. 2-4. Stirling Cycle Refrigerator Ideal Pressure-Volume and Temperature-Entropy Diagrams

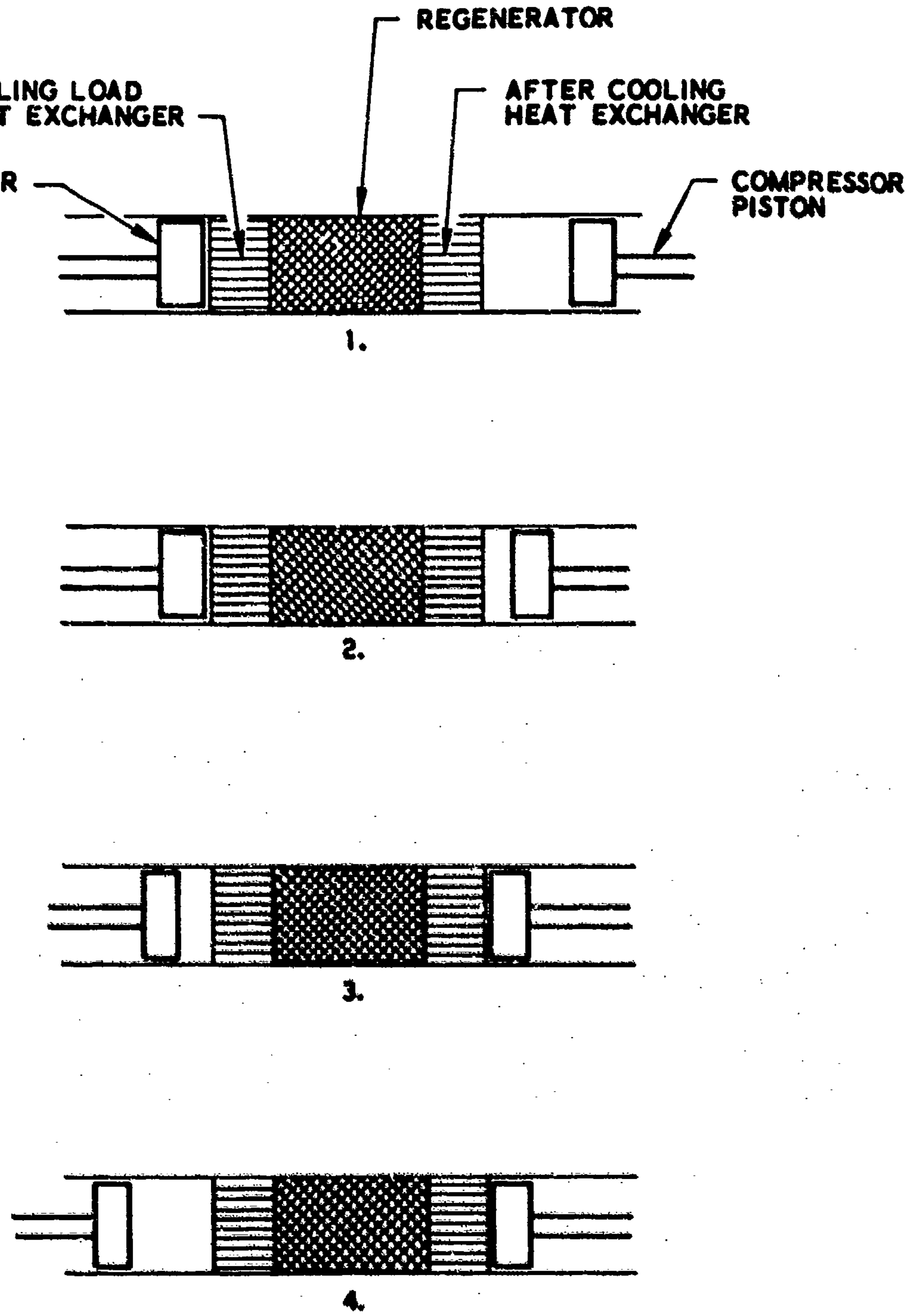


Fig. 2-5. The Practical Stirling Refrigerator (Ref. 16)

incremental displacement of both pistons. During this transfer, the lost energy of expansion is replaced in the cooling load heat exchanger, and the temperature is raised to the ambient temperature in the regenerator.

2. PRODUCTION AND DEVELOPMENT OF STIRLING CYCLE REFRIGERATORS

The following four companies are engaged in development and/or production of Stirling cycle refrigerators: Philips Laboratories, Malaker Corporation, Hughes Aircraft Company, and Kinergetics.

Philips, the pioneer in development of Stirling cycle refrigerators, built their first machine in 1954 for the purpose of air liquefaction. Presently, Philips produces a variety of machines for laboratory and industrial use as well as miniature units for aircraft use. Two miniature flight units designated "Cryogem" provide cooling in the range of 20 to 40 K (2.0 W at 30 K) and are used for cooling IR detectors in aircraft. Data on two additional units are provided in Table 2-3.

Malaker Corporation has been engaged in the development of refrigeration equipment since the 1950's. The majority of their research and development has gone into the Stirling cycle and has been concentrated on small units. Malaker has produced a number of units with very high thermal efficiencies and is actively engaged in additional development and modification of their units. Recent work has been devoted to the modification of one of their units (ID #33) to make it adaptable to space operation. The primary modification consists of providing an all-welded case around the units to allow larger temperature excursion during operation without freezing up the existing O-ring seals and leaking the working gas. Some additional modifications have been made on the above unit in order to meet launch vibration requirements. The current guaranteed operating life (maintenance-free) is 1000 hr. Two of these units presently are operating successfully on a spacecraft. A large variety of units has been available ranging from 2.0 W at 25 K up to 110 W at 77 K. However, at this time, all operations of the Malaker Corporation have been terminated.

Table 2-3. Identification of Stirling Cycle Refrigerators

Identification Number	Manufacturer or Developer	Model, Description, or Program	Operating Temperature Range (K)	Typical Refrigeration Performance	Status	Power: Input (W)	Power: Input/Refrig. (W/W)	Weight: Total (lb)	Weight: Tot/Refrig (lb/W)	Remarks	Ref.
29	Malaker	VII-C Standard	25 to 77	2.0 W at 25 K 15.0 W at 77 K	Production	480 295	240 26.4	15.5	7.75 1.03		17
36	Philips	Cryogem 42100	20 to 40	2.0 W at 30 K	Production	350	175	12	6.0	Aircraft Usage	9
37	Philips	Cryogem 42151	20 to 40	2.0 W at 30 K	Production	550	275	25	12.5	Aircraft Usage	9
30	Malaker	Mark VII-R	40 to 125	60.0 W at 77 K	Production	1220	20.5	40	0.67		17
32	Malaker	Mark XIV-A	45 to 100	1.5 W at 60 K 2.8 W at 77 K 4.5 W at 100 K	Production	120 108 100	80.0 38.5 22.0	5.5	3.7 1.97 1.22		17
34	Philips	----------	7 - 300	0.50 W at 12 K	Prototype	700	1400	35	70		9
24	Hughes	----------	~25	0.80 W at 25 K	Prototype	620	775	16	20		4
27	Hughes	----------	~80	15 W at 80 K	Prototype	500	33.5	10	0.67		4
28	Hughes	----------	~80	2.0 W at 77 K	Prototype	580	290	11	6.5		4
47	Kinergetics[a]	SRS-07 (Two Stage)	50 - 77	0.50 W at 60 K 1.0 W at 77 K	Production	40	80 40	4.7	9.4 4.7		18
38	Philips	Micro Cryogem	40 - 300	1.5 W at 77 K	Production	90	60	3	2.0		9
31	Malaker	Mark XX	40 - 120	110 W at 77 K	Production	1990	18.1	65	0.59		17
33	Malaker	Mark XV	54 - 100	1.0 W at 77 K	Production	29.5	29.5	5	5.0		17
39	Philips (Netherlands)	X-20	12 - 300	10 W at 20 K	Prototype	1750	175	112	11.2		16
52	Philips	P/N 460600	50 - 80	1.0 W at 50 K	Prototype	120	120	4	4.0		19
73	Malaker	Mark XVII-1	- 77	4.3 W at 77 K	Production	280	65.3	13	3.0		17
74	Malaker	Mark XVI-3	77 - 110	8.3 W at 77 K	Production	208	25.0	10	1.2		17
75	Malaker	Mark XV-4	54 - 100	1.0 W at 77 K	Production	29.5	29.5	5	50		17

[a]Formerly Submarine Systems

Hughes has developed several Stirling refrigerators primarily for use in aircraft. These units are not commercially available but essentially provide a support function for in-house activities. These units provide cooling in the 25 to 77 K range.

Kinergetics currently has no active development program in Stirling units. The unit listed in Table 2-3 (ID #47) is no longer in production.

3. PERFORMANCE DATA

Performance data are summarized in Table 2-3 for a number of typical production units and various prototype or developmental units. Additional details for each unit are provided in Table 2-14. The refrigeration specific weight and power versus temperature for each unit are plotted in Fig. 2-6. As seen from Fig. 2-6, the data points are relatively well grouped together, and the general trend illustrates (when compared with data for other systems) that the Stirling cycle units possess one of the best weight and power characteristics of refrigerators applicable to spaceborne operation.

At first glance, a number of data points in Fig. 2-6 may appear questionable and require further clarification. For example, unit #28 has essentially the same specific power ratio as #37 at a much higher temperature. Unit #28 involves a remote detector location with associated heat losses, and as a result the actual refrigeration being produced is higher than the net cooling load used to compute the specific power ratio. Therefore, the power ratio is higher.

In other cases, where a system provides multiple stages of cooling, the specific weight and power is computed for each stage and entered in the chart (this accounts for more than one data point for specific units such as #29, #32, and #47). As a result, the specific weight and power ratio would not necessarily correlate satisfactorily with other data for single stage cooling systems.

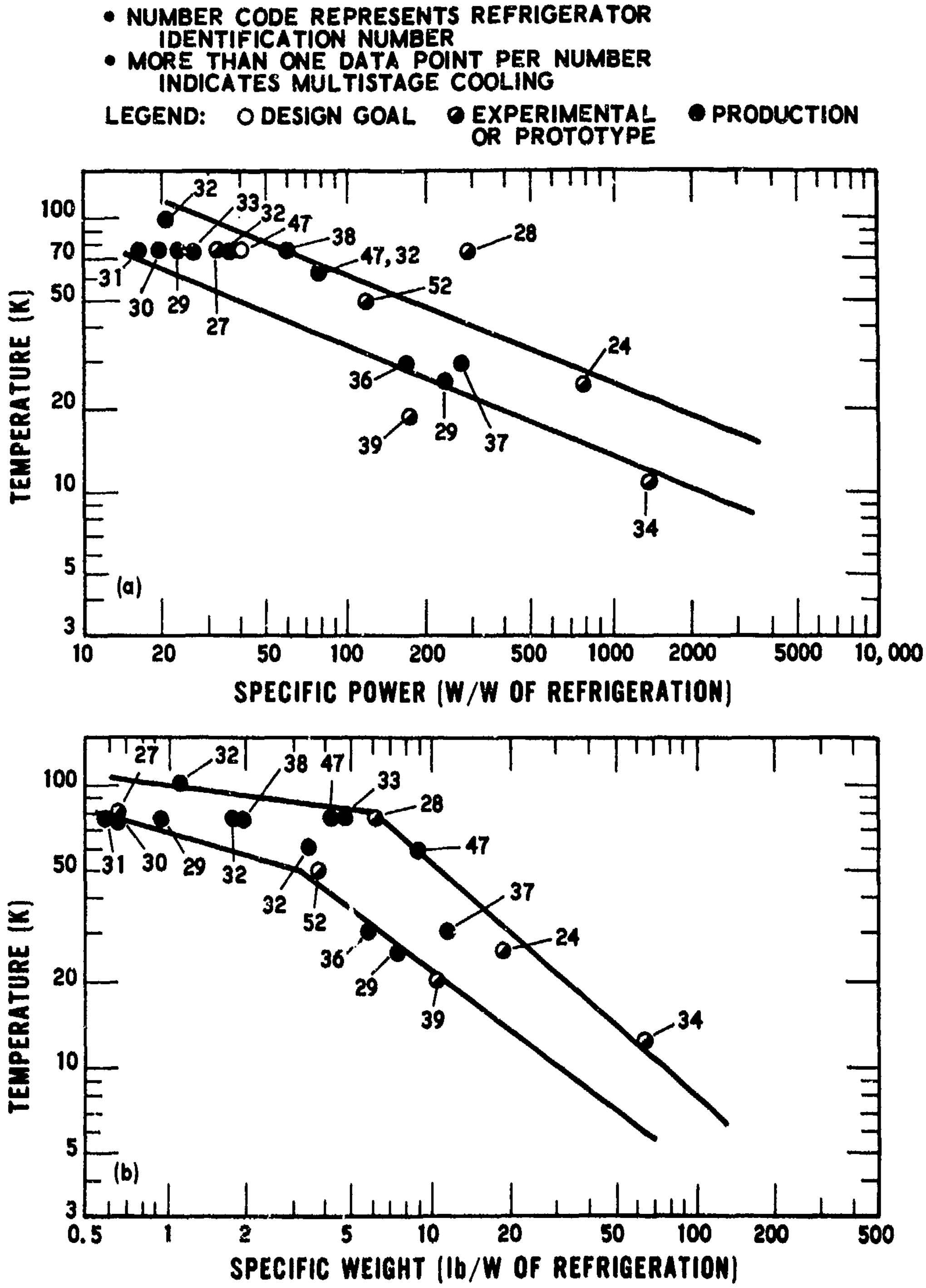


Fig. 2-6. Stirling Cycle Refrigerator Power and Weight Characteristics

4. FUTURE DEVELOPMENT AND PROBLEM AREAS

Previous development and design requirements for Stirling cycle refrigerators have been oriented primarily towards minimum weight, volume, and power, whereas extended life has not been a high priority objective. Most of the production units manufactured by Malaker and Philips have maintenance-free operating lives of 500 to 1000 hr. As a result, existing hardware does not possess the lifetime characteristics desirable for extended spaceborne operations but, as previously indicated, can be modified for limited spaceborne operations.

The potential problem areas associated with the Stirling refrigerator have been evaluated by M. Bello (Ref. 3) and are summarized in Table 2-4. Traditionally, Stirling refrigerators have been limited for three reasons: (a) seal wear causing working fluid leakage, (b) limited life of dry lubricated bearings, and (c) excessive vibration. Recent developments which would eliminate the above problem areas are: (a) utilization of rolling seals thereby allowing the use of liquid lubricants and reducing helium losses and (b) the rhombic drive assembly which reduces vibration and bearing loads. The rolling seals have been tested in miniaturized configurations for approximately 11,000 hr at 1800 rpm without evidence of wear. It is concluded in the above evaluation that the development of a compact Stirling refrigerator capable of operating continuously for 20,000 hr in a spacecraft environment is feasible.

E. GIFFORD-McMAHON/SOLVAY CYCLE REFRIGERATORS

1. BACKGROUND

The practical Stirling and VM refrigerators achieve compression, expansion, and heat transfer processes in a single mechanical unit. However, refrigerators can be built which use regenerative exchangers in which the compression, expansion, and heat exchange components are separated. In recent years, this split component system has gained a great deal of popularity.

Table 2-4. Stirling Refrigerator Problem Areas (Ref. 3)

Components	Problem Areas	State of Development	Number of Components
1. Regenerator	Degradation characteristics due to aging and contamination require further assessment-organic spacer material	2	4
2. Counterflow Heat Exchanger	Manufacturing tolerances may shift flow characteristics	1	1
3. Displacer Seals*	Wear of displacer seals causes both contamination and leakage	4	2
4. Piston Rod Seal*	Use of rolling seals will require development of working fluid resupply due to leakage through the polymeric seal material	4	3
5. Rotary Motor	Needs life test to verify that wear is not a problem	1	1
6. Pistons (Displacers)	Large temperature gradient along the length of pistons requires special material	2	2
7. Rhombic Drive Bearings	Requires oil lubrication for extensive life	2	6
8. Linkages in Drive Assembly	Design factor	2	10

*Currently life limiting, correctable by development.

Status Key:
- 0 Fully developed
- 1 Requires life test
- 2 Performance testing
- 3 Currently being developed
- 4 Requires development
- 5 Beyond state of the art (1975)

By separating the expander from the compressor, it is possible to construct a system consisting of a simple, lightweight, compact cooling unit, which can be more easily integrated with the load, and a compressor which can be located separately. The compressor is then connected to the expander with long flexible lines carrying the high and low pressure working fluid. Because of this characteristic and the commercial attractiveness of this type of system, there are many varieties on the market. The systems are basically the same in that nearly all use hermetically sealed compressors so that the system variations are primarily confined to methods of operating the expander unit and various design, material and manufacturing techniques to produce more reliable, long-life, low-cost systems.

2. OPERATION AND CYCLE DESCRIPTION

The basic refrigeration cycle used in this type of system was originally conceived by Ernest Solvay in 1886 as a basic derivative of the Stirling cycle. A number of modifications have been made by various researchers such as K. W. Taconis, W. E. Gifford, and H. O. McMahon. Refrigerator units manufactured are usually marketed using various names as Gifford-McMahon and Solvay, with and without the adjective "modified." They are basically the same cycle but with different expander modifications.

The basic Solvay expansion process is illustrated in Fig. 2-7. In position 1 the inlet valve is open and the exhaust closed. The regenerator and other void volumes are filled to the higher pressure. From 1 to 2 the piston moves outward and working fluid enters the cylinder after being cooled in the regenerator. At point 2 the inlet valve is closed and the fluid pressure falls until the piston reaches its outermost position. At position 3 the exhaust valve is opened and the fluid in the system expands to 4. From 4 to 5 the piston moves inward, expelling the cold working fluid from the system after being warmed in the regenerator. At 5 the exhaust valve is closed and the piston continues to move until it reaches the innermost position at 6. At position 6 the inlet valve is opened and the fluid in the system is compressed from 6 to 1.

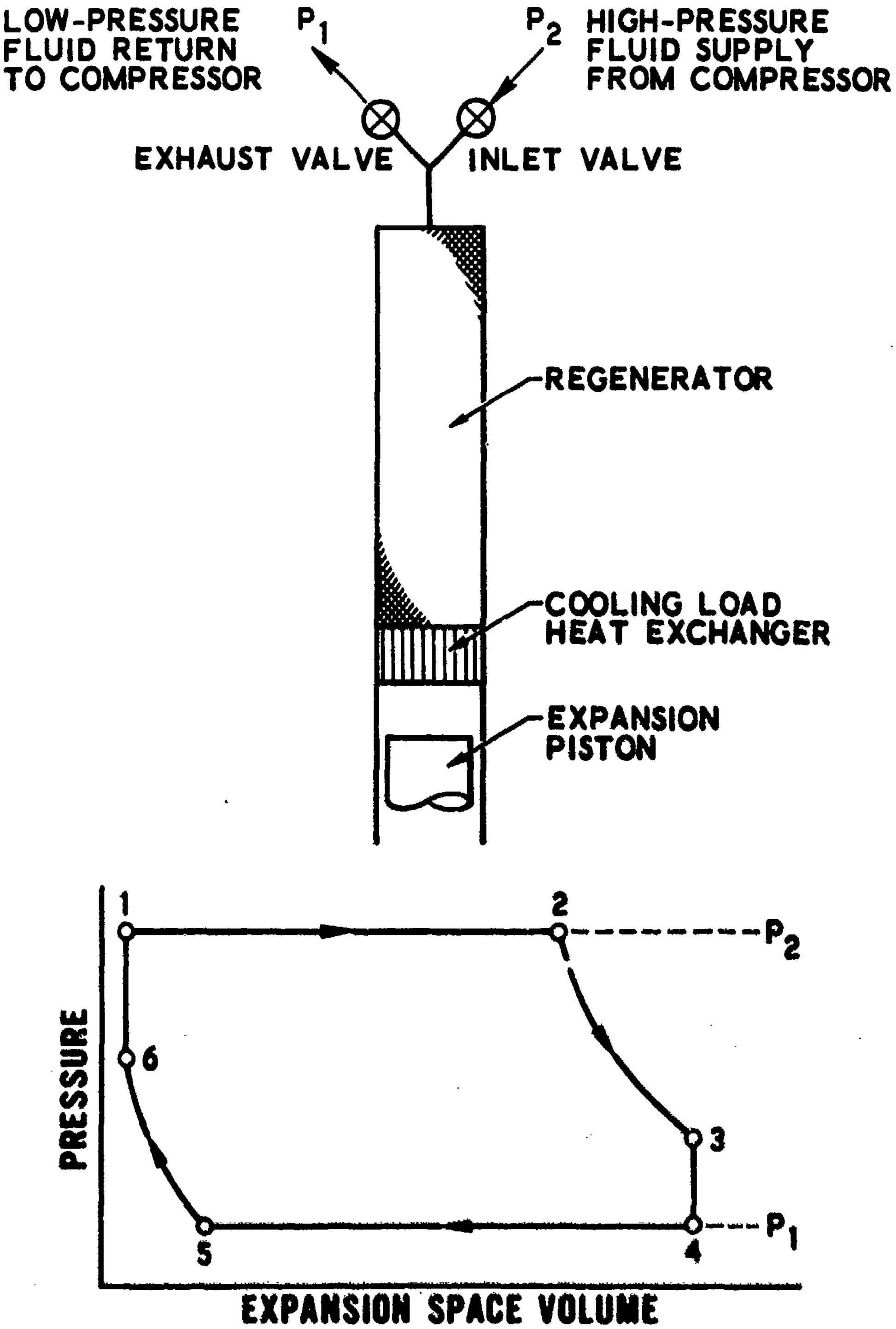


Fig. 2-7. Solvay Refrigeration Cycle

3. PERFORMANCE DATA

A brief summary of refrigeration performance and weight and power data for the various units in this classification is shown in Table 2-5. System weight and power input requirements per watt of refrigeration as a function of temperature are plotted in Fig. 2-8. The data points for both weight and power fall within a relatively narrow band. Additional details of the individual refrigerator units are given on Table 2-14 listed in order of the identification number. The weight and power requirements are substantially higher than for comparative Stirling cycle units, and it is not likely that any significant improvement in power requirements will be made. The most significant portion of the actual hardware weight is primarily due to the compressors (which are normally modified airborne freon units). Developments for use in other cycles such as rotary compressors using gas bearings and rotary-reciprocating machinery described in Section II-F could provide a significant reduction in compressor weights.

4. MANUFACTURERS

The primary manufacturers of lightweight compact units of this type using variations of the Solvay cycle include Cryogenic Technology, Inc., Cryomech Inc., and Air Products and Chemicals. A variety of units have been manufactured for cooling applications such as: (1) general purpose laboratory use, (2) IR detectors, lasers and miscellaneous electrooptical devices for both ground and airborne operation, (3) storage of biological samples, and (4) material studies at low temperature.

a. Cryogenic Technology, Inc. (CTI)

CTI, which evolved from A. D. Little, as been involved with much of the early development work in units of this type. CTI produces a number of units under the "Cryodyne" trademark which provide cooling in the range of approximately 4 to 150 K in capacities ranging from 1 to 125 W. Most of these units have been designed for industrial or laboratory use where weight

Table 2-5. Identification of Gifford-McMahon/Solvay Cycle Refrigerators

Identification Number	Manufacturer or Developer	Model, Description, or Program	Operating Temperature Range (K)	Typical Refrigeration Performance	Status	Power: Input (W)	Power: Input/Refrig. (W/W)	Weight: Total (lb)	Weight: Tot/Refrig (lb/W)	Remarks	Ref.
4	Cryogenic Technology	Model 400	3.8 to 4.5	1.25 W at 4.2 K	Production	5500	4400	925	740	G-M + J-T	9
5	Cryogenic Technology	Model 0110	6.5 to 25	1.0 W at 10 K	Production	2100	2100	205	205	G-M	20
11	Cryomech Inc.	Model GB02	7.5 to 25	1.0 W at 9.5 K	Production	3000	3000	200	200	G-M	9
12	Cryomech Inc.	Model GB12	9 to 30	4.0 W at 13 K	Production	3000	750	200	50	G-M	16
6	Cryogenic Technology	Model 1023	10 to 28	2.0 W at 13 K	Production	6100	3050	458	229	G-M	9
7	Cryogenic Technology	Model 350	15 to 28	2.0 W at 18.5 K (+5.0 W at 77 K)	Production	2100	1050	197	98.5	G-M	20
8	Cryogenic Technology	Model 0120/PC-30	19 to 28	1.0 W at 26 K	Production	650	650	25.5	25.5	G-M	9
9	Cryogenic Technology	Model 0125	19 to 28	1.0 W at 26 K	Production	750	750	60	60.0	G-M	20
10	Cryogenic Technology	----------	25 to 150	125 W at 77 K	Production	6100	48.7	143	1.14	G-M	9
13	Cryomech Inc.	Model AL01	23 to 80	1.0 W at 25 K	Production			----	------	G-M	9
14	Cryomech Inc.	Model AL02	23 to 89	10.0 W at 30 K	Production	3000	300	200	20	G-M	9
2	Air Products	Model I-1	30 to 150	20.0 W at 77 K	Production	1700	85	140	7	Solvay	9
45	Air Products	Military Application	77	1.5 W at 77 K	Production	340	226	11	7.3	Solvay	21
54	Air Products	CS-1003	50 to 300	1.0 W at 77 K	Production	400	400	63	63	Solvay	22
46	Kinergetics*	SRS-07	50 to 77	1.0 W at 38 K 2.0 W at 50 K	Production	400	400 200	12.4	12.4 6.2	Solvay	18
50	Air Products	CS-102	30 to 200	17.0 W at 77 K	Production	1735	102	150	8.75	Solvay	21
51	Air Products	CS-202	12 to 300	1.0 W at 17 K	Production	1735	1735	150	150	Solvay	21
61	Air Products	MS-1003	30 to 77	1.0 W at 77 K	Production	368	368	14.5	14.5	Developed for airborne applications	21
66	Cryogenic Technology	0120	19 to 30	1.0 W at 26 K	Production	800	800	25	25.0	Designed for airborne use (MIL-E-5400)	23
67	Cryogenic Technology	0277	40 to 120	3.0 W at 77 K	Production	525	175	16	5.33	Designed for airborne use	23
68	Cryogenic Technology	1020	13 to 20	10 W at 20 K (+10 W at 77 K)	Production	5.6kw	560	458	45.8		23

*Formerly Submarine Systems

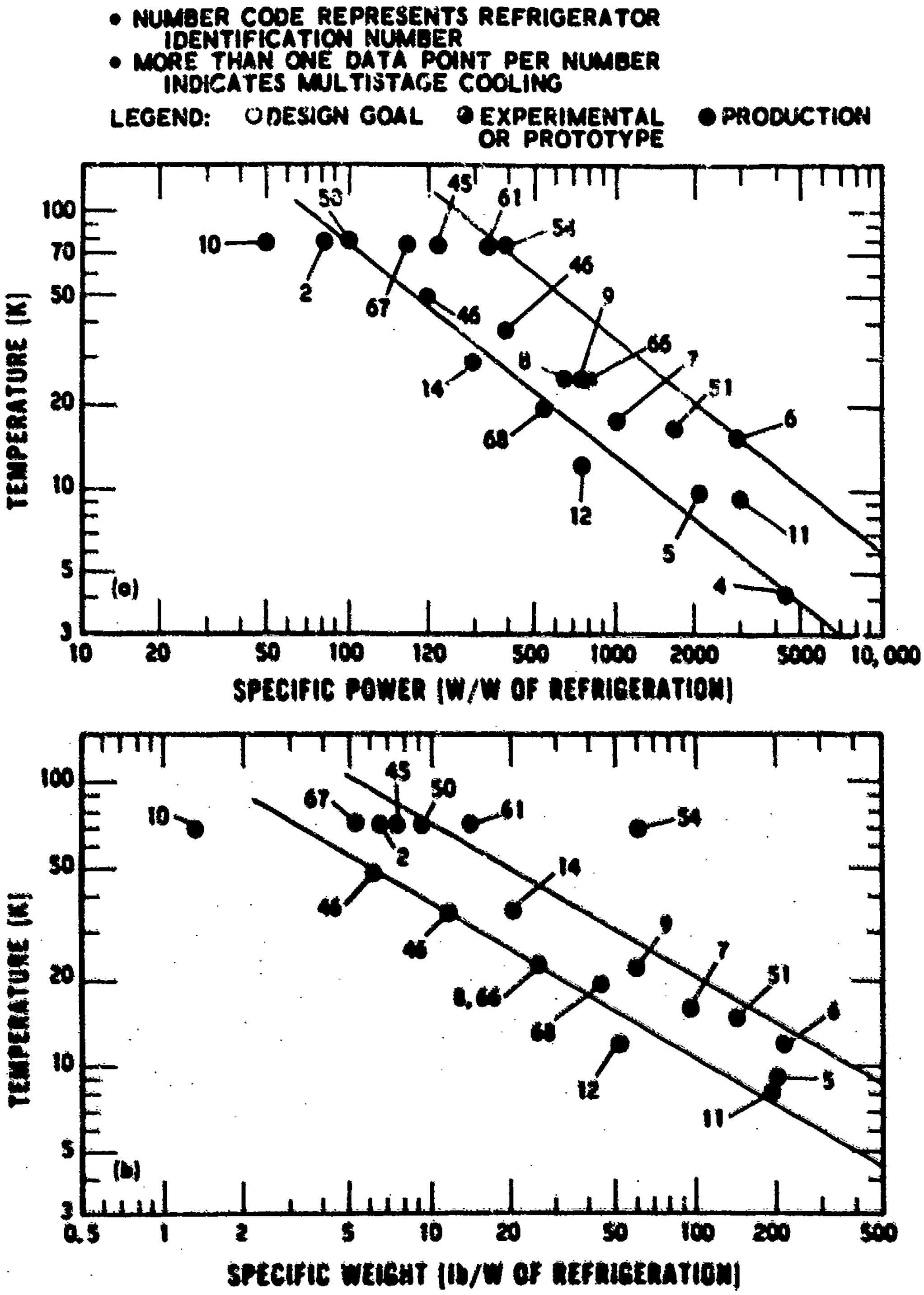


Fig. 2-8. Gifford-McMahon/Solvay Cycle Refrigerator Power and Weight Characteristics

and power are of low priority. As a result, the units utilize power compressors which are relatively heavy and have high power consumption characteristics. However, units have been designed for use in airborne IR detector systems (including models 0120 and 0277 listed as ID #66 and 67, respectively). Several variations of these units are also available by interchanging compressors. Both of the above units are compact, lightweight units which provide approximately 1000 hr of maintenance-free operation. The model 0120 produces cooling in the 19 to 30 K range (1.0 W at 26 K) while the model 0277 provides cooling in the 40 to 120 K range but was designed primarily to produce 3.0 W at 77 K. Both the 0120 and 0277 models are designed to meet military airborne requirements (MIL-E-5400), but only the 0120 model has been qualified.

b. Cryomech, Inc.

Cryomech is a small company which was organized in 1964 by Dr. W.E. Gifford who did much of the early development of the system which bears his name. Cryomech makes a standard line of a number of Gifford-McMahon units primarily for industrial and laboratory use. The units listed in this report provide cooling in the range of 7.5 to 89 K in capacities ranging from 1 to 20 W. Most of these units use oil-lubricated freon compressors. Although the specific power and weight requirements become substantial at these lower temperatures in the 10 K region (as indicated by Fig. 2-8 and Table 2-5), the remote cooling head permits utilization; otherwise it might be impractical.

c. Air Products and Chemicals

Air Products and Chemicals produces a number of units designed primarily for medical, laboratory and industrial or scientific uses in a broad temperature range of 12 to 300 K with capacities from 1 to 20 W. Most of these units which are commercially available utilize oil-lubricated compressors. However, a recently released model (MS-1003, ID #61) utilizes a dry-lubricated compressor. These units are very compact, weigh less than 15 lb, and have guaranteed operating lives in the order of 10,000 hr when maintenance is provided at 1200-hr intervals.

5. DEVELOPMENT POTENTIAL

Because of the commercial attractiveness of the Gifford-McMahon/ Solvay cycle units, substantial development effort and production knowledge have been achieved. As mentioned previously, the primary advantage is that the cooling head can be separated from the compressor.

The primary limitation is that the lower efficiency, as compared with VM or Stirling systems, generally requires significantly higher power input than the latter two cycles for the same cooling load. Nevertheless, these units currently provide the bulk of commercial low temperature cooling and provide the longest unattended lifetime. Although there is not likely to be significant improvements in power requirements, substantial weight reductions in the systems may be expected with the use of compressor units which are optimized for minimum weight.

F. BRAYTON AND CLAUDE CYCLE SYSTEMS

1. TURBOMACHINERY REFRIGERATION SYSTEMS

a. Background

In recent years efforts have been directed toward developing gas-bearing supported turbomachinery suitable for use in small cryogenic refrigerators. Two refrigeration cycles suitable for this application are the reversed Brayton cycle and the Claude cycle.

Turborefrigerators employing gas-bearing turbomachinery have the potential for high reliability and long, maintenance-free life. Lubrication of the bearings with the cycle working fluid excludes lubricants as a source of contamination and fouling in the low temperature regions of the cycle. The absence of continuously rubbing parts eliminates the wearing mode of failure that is typical of non-lubricated or dry-lubricated machinery. Hence, with gas-bearing turbomachinery there is the expectation for long life, probably limited by the number of start-stop cycles rather than hours of operation. The performance of miniature turbomachinery refrigerators is a strong

function of their capacity. This is primarily because the working fluid flow rates in small refrigerators are substantially lower than the normal range appropriate to turbomachinery. As capacity and cycle flow are increased, the turbomachinery design requirements move in a more favorable direction and the relative size, weight, and performance of the refrigerator are substantially improved.

b. Description and Operation of Brayton and Claude Cycles

In the reversed Brayton cycle, the compressor operating at ambient temperature compresses the cycle gas, and heat is rejected to ambient temperature. The high-pressure gas is then passed through a series of counterflow cryogenic heat exchangers and is expanded across one or more turbines where the energy is extracted. The gas is then directed through the refrigeration load. The Claude cycle is similar except that an additional heat exchanger and J-T valve are added at the cold end to achieve a further reduction in the temperature.

The thermodynamic processes associated with each of the two cycles are shown in Figs. 2-9 and 2-10 for single stage systems. In the Brayton cycle gas is compressed with some increase in entropy from 1 to 2. The heat of compression is rejected to the ambient temperature heat sink in an after-cooler from 2 to 3. The high-pressure fluid is cooled from 3 to 4 in the main heat exchanger. The pressure at 4 is slightly less than at 2 due to the flow losses in the two heat exchangers. The fluid is expanded from 4 to 5 with some entropy increase, and is then warmed to 6 by passage through the load heat exchanger. The fluid is warmed from 6 to 1 in the main heat exchanger as it returns to the inlet side of the compressor.

As the operating temperature of the Brayton refrigerator is lowered, point 5 (Fig. 2-9) will enter the two-phase region of the working fluid, and the fluid will leave the expander as a two-phase mixture. For refrigeration at temperatures within the two-phase region of the working fluid, it has become accepted practice to perform the expansion process through a throttle

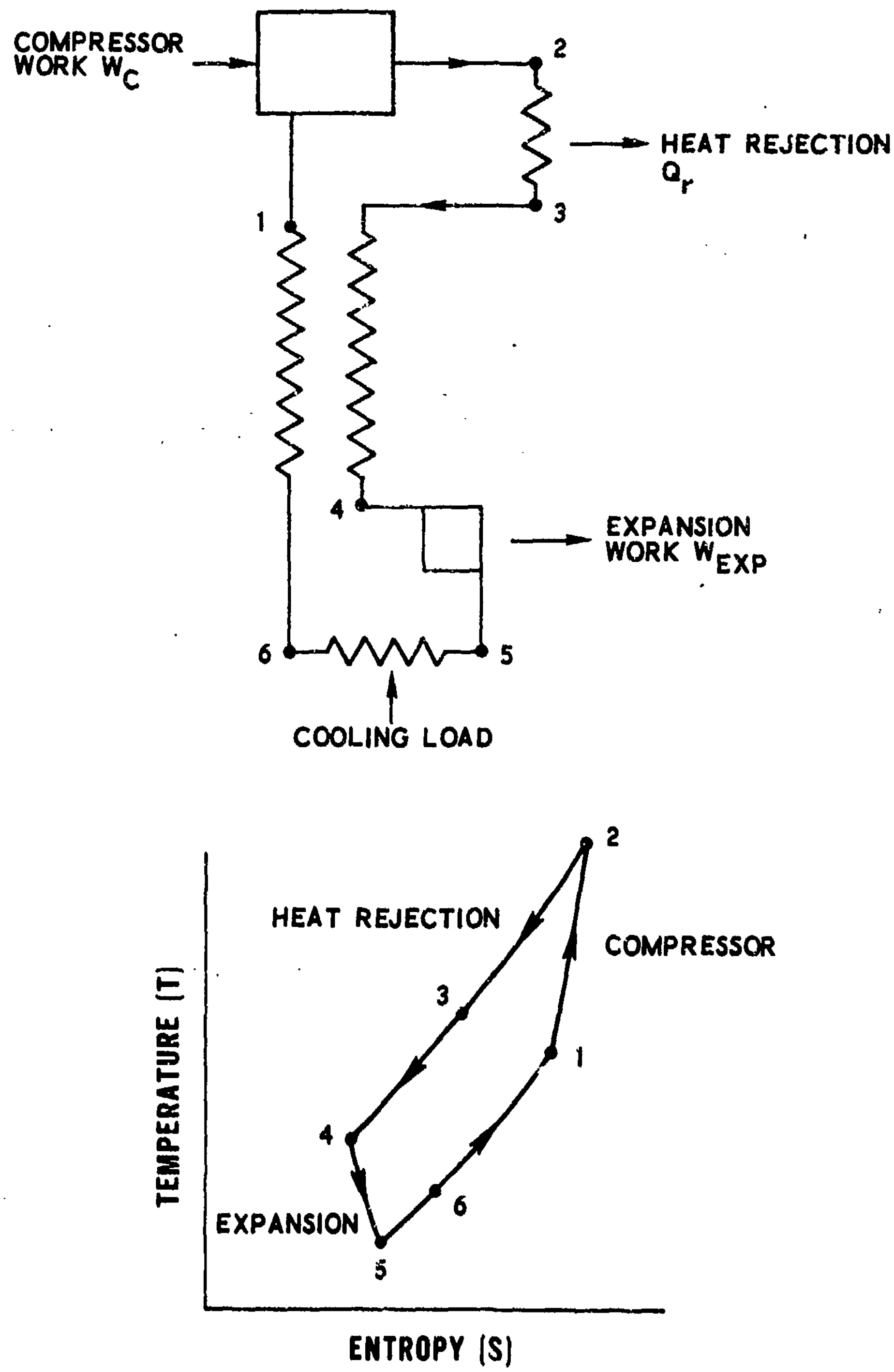


Fig. 2-9. Reversed Brayton Refrigeration Cycle

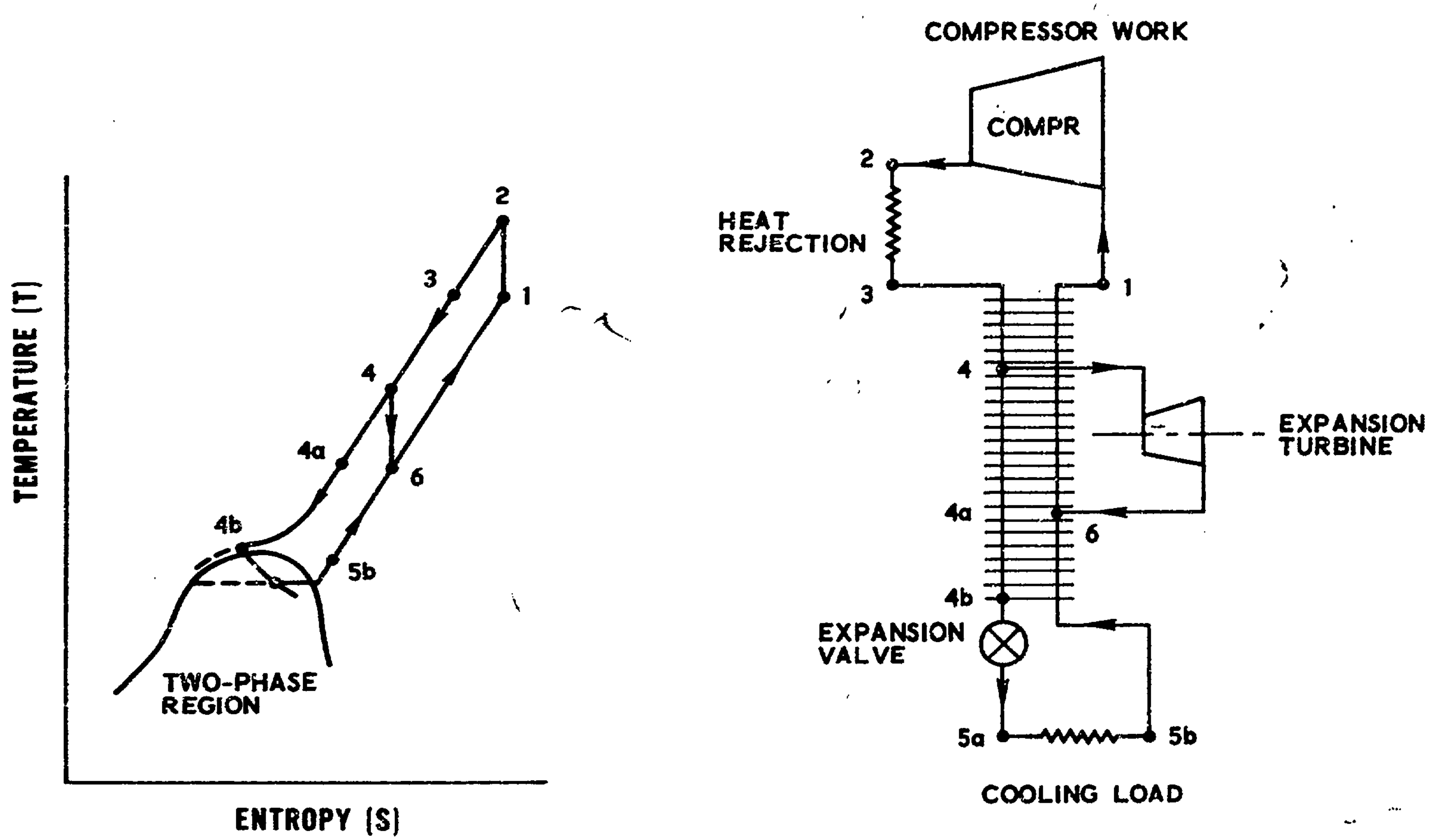


Fig. 2-10. Claude Refrigeration Cycle

valve, as in the J-T cycle, rather than in an expansion engine. It is seen from Fig. 2-10 that the Claude cycle is effectively a J-T cycle in which the effective sink temperature is lowered by a Brayton cycle refrigerator. Typical turbomachinery systems utilizing the Brayton and Claude cycles are shown in Fig. 2-11.

c. Turbomachinery Development Program

Companies that have engaged in the development of turbomachinery refrigeration systems utilizing the Brayton or Claude cycle are the General Electric Company (Schenectady, New York), Garrett-AiResearch Company, A. D. Little (Cambridge, Mass.), Linde Division of Union Carbide (Cryogenic Products Division, Tanawanda, New York), and the Hymatic Engineering Company of England.

(1) General Electric

The General Electric Company has contracts with both the U.S. Army Mobile Equipment Research Development Center (MERDC) and the Air Force Flight Dynamics Laboratory. The Army program, as summarized in Table 2-6, calls for development and delivery of a 2 W, 4.4 K turbomachinery refrigerator. This system utilizes helium in a Claude cycle with an estimated power input of approximately 9 kW, an estimated weight of about 100 lb, and a projected continuous operating time of 10,000 hr. This type of system is intended for use with mobile ground-based cryogenic temperature electrical generating equipment. General Electric has completed the design and fabrication and has carried out extensive tests on a turboexpander (alternator loaded), heat exchangers, and turbocompressors. They are using a dynamic gas-bearing technique and have made considerable progress in heat exchanger design.

The Air Force program involves development of two refrigerators. One refrigerator has three simultaneous cooling loads of 3.5 W at 5 K, 40 W at 50 K, and 200 W at 150 K. Refrigerator input power and weight are 16 kW and 250 lb, respectively. The other refrigerator has two simultaneous

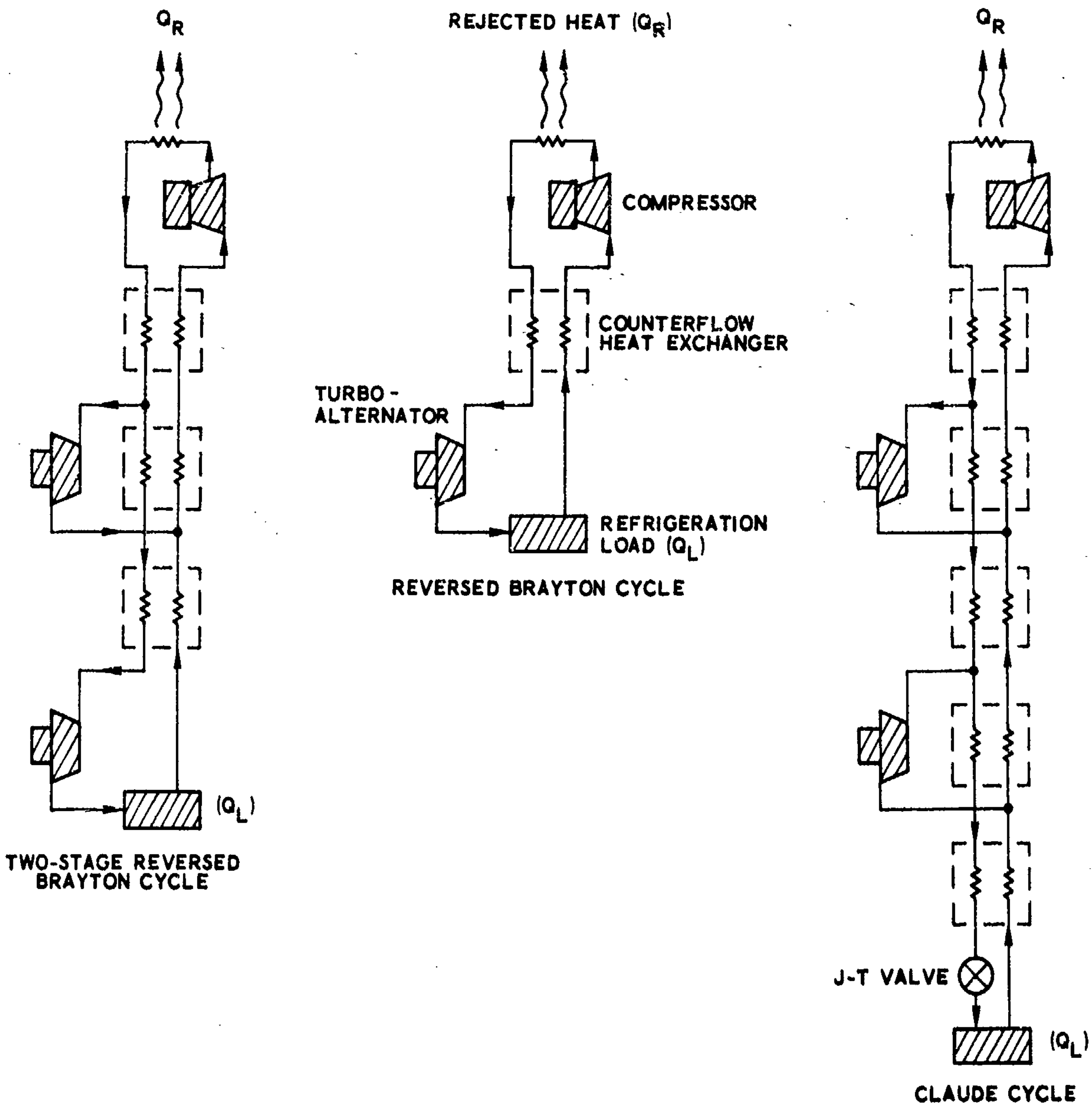


Fig. 2-11. Reversed Brayton and Claude Cycle Turbomachinery Refrigerator System Schematics

Table 2-6. U.S. Army MERDC* Advanced Technology Program Cryogenic Turbo Refrigerator System

Program Elements

Cycle Analysis	Refrigerator Sizing Studies
Advanced Component Development	Turboalternator, Compressor
Laboratory System Development	80 K Refrigerator 4.4 K Refrigerator
Advanced Refrigerator Design	Superconducting Electrical Machinery

Performance Goals and Requirements

Refrigerator Performance

Load Temperature	4.4 K
Load Capacity	2.0 W
Cycle Fluid	Helium

Turboalternators

Inlet Temperature, K	75	12
Electrical Power Output, W	32.87	8.85
Speed, rpm	205,200	78,900

Compressor

Number of Stages	3
Overall Pressure Ratio	2.52
Speed	90,000 rpm

Cryogenic Heat Exchangers

Exchanger	A	B	C	D	E
Effectiveness	0.990	0.936	0.990	0.936	0.990
Core Weight, Lb	10.65	0.17	5.28	0.15	0.99

*Mobile Equipment Research and Development Center

loads of 1.5 W at 12 K and 40 W at 60 K. This refrigerator requires 4 kW power and an estimated weight of 250 lb. General Electric is obligated to perform 5000-hr endurance tests on both refrigerators prior to delivery to AFFDL. The program requirements are summarized in Table 2-7.

(2) Garrett-AiResearch

AiResearch has had a number of contracts with both the Air Force and Army for development of miniature turbomachinery systems. The only contract which resulted in development of a prototype unit was terminated in 1967. The objective of this program was to develop a refrigerator with a cooling capacity of 2 W at approximately 77 K which would be suitable for use with spaceborne IR sensor devices. The identification number for this unit is 58.

(3) A. D. Little

A study to develop a refrigerator to produce 1 W at 3.6 K for space applications was conducted by A. D. Little for AFFDL in 1968. The major components of the system were built and tested; however, a complete working model was not constructed. Available data on this unit (ID #60) is taken from Ref. 16.

(4) Hymatic

As reported in Ref. 16, Hymatic Engineering Company has developed a prototype Brayton cycle unit which produces 0.30 W at 28 K. Other available information is listed in Tables 2-8 and 2-14 under ID #59.

(5) Air Products

Air Products is apparently not active in the development of turbomachinery refrigeration systems. Data were obtained from Ref. 9 for a system developed for laboratory operations (ID #1).

Table 2-7. U.S. Air Force FDL Advanced Development Program Cryogenic Turbo Refrigerator Systems

Program Elements

Optimization Analysis	Reliability, Power Input, Weight, Size, Cost
Design	30,000-Hr Life
Fabrication	Develop Components
Test	Performance, Environmental, 5000-Hr Endurance
Deliver	Install, Operate, Instruct
Value Analysis	Product Improvement

Performance Goals and Requirements

	Refrigerator A (# 48)[§]	Refrigerator B (# 49)
Load at Temperature	3.5 W at 5 K ± 0.05 K 40.0 W at 50 K ± 0.5 K 200.0 W at 150 K ± 2.0 K	1.5 W at 12 K ± 0.1 K 40.0 W at 60 K ± 0.5 K
Input Power	16 kW	4 kW
Weight	250 Lb	100 Lb[*]
Volume	25 in. dia. x 48 in. cyl.	20 in. dia. x 65 in. cyl.
Load Volumes at Temperature	6 x 6 x 6 in. at 5 K 3 x 6 x 6 in. at 50 K 3 x 6 x 6 in. at 150 K	Unspecified at 12 K Unspecified at 60 K
Compressor Drive	Electric Motor	Magnetic Coupling/Space Power Drive

[*] Latest Estimate = 250 lb

[§] This program was recently terminated.

Table 2-8. Identification of Turbomachinery Refrigerators

Identification Number	Manufacturer or Developer	Model, Description, or Program	Operating Temperature Range (K)	Typical Refrigeration Performance	Status	Power		Weight		Remarks	Ref.
						Input (kW)	Input/Refrig. (W/W)	Total (lb)	Tot/Refrig (lb/W)		
47	General Electric	Army MERDC[c]	4.4 - 80	2.0 W at 4.4 K	Development	9.0	4500	~100	50		25, 26
48	General Electric	AFFDL ADP (A)	3.5 - 150	3.5 W at 5 K 40.0 W at 50 K 200.0 W at 150 K	Development	16.0		~250	7.15 6.25 1.25	Three stages of cooling	26
49	General Electric	AFFDL ADP (B)	12 - 30	1.5 W at 12 K 40.0 W at 60 K	Development	4.0	2660 100	~250	167 6.25	Two stages of cooling	26
1	Air Products	E-311	3.3 - 4.5	1.0 W at 3.9 K	Production	7.0	7000	420	420		9
58	Garrett AiResearch	AFFDL Development	80	2.0 W at 80 K	Prototype	375W	188	151	75.5		16
59	Hymatic	----------	19 - 28	0.30 W at 28 K	Prototype	----	----	----	------		16
60	A.D. Little	AFFDL Development	3.6	1.0 W at 3.6 K	Prototype	1310W	1310	124	124		16

[c] Mobile Equipment Research and Development Center

This program was recently terminated.

d. Performance Data

The data available on these units are summarized in Table 2-8 with specific weight and power characteristics shown in Fig. 2-12. Additional data on available characteristics are provided in the complete listing of mechanical refrigeration in Table 2-14 at the end of this section. Included in Fig. 2-12 is a shaded band representing results of analytical studies conducted by General Electric. Design characteristics resulting from one such study are illustrated in Table 2-9. Based on several similar studies by General Electric, power and weight estimates for turbomachinery refrigerators designed to operate at 4.4 K and 20 K are shown in Fig. 2-13.

e. Development Potential

On the basis of recent evaluations made by M. Bello (Refs. 3 and 24), the turbomachinery refrigerator has a favorable potential for achieving long life spaceborne systems, and there are currently no identifiable life limiting components. Potential problem areas identified in Ref. 3 are shown in Table 2-10.

In summary, the turbomachinery developments in miniature refrigerators seem to indicate that (1) it is possible to construct small turboexpanders and compressors suitable for refrigeration application, (2) the reliability and continuous running time for these units can be very good--with maintenance intervals of 10,000 hr or better expected, (3) the heat exchanger and compressor presently set the limit on size reduction of turborefrigerators, and (4) it is not likely that a miniature turbomachinery refrigerator will be commercially available for at least several years.

2. ROTARY-RECIPROCATING (RR) REFRIGERATOR SYSTEMS UTILIZING THE BRAYTON CYCLE

a. Background and Description

Miniature refrigerators utilizing reciprocating machinery in the Brayton cycle have recently been developed for space applications (Ref. 27). To achieve the requisite reliability, a novel approach has been used in the

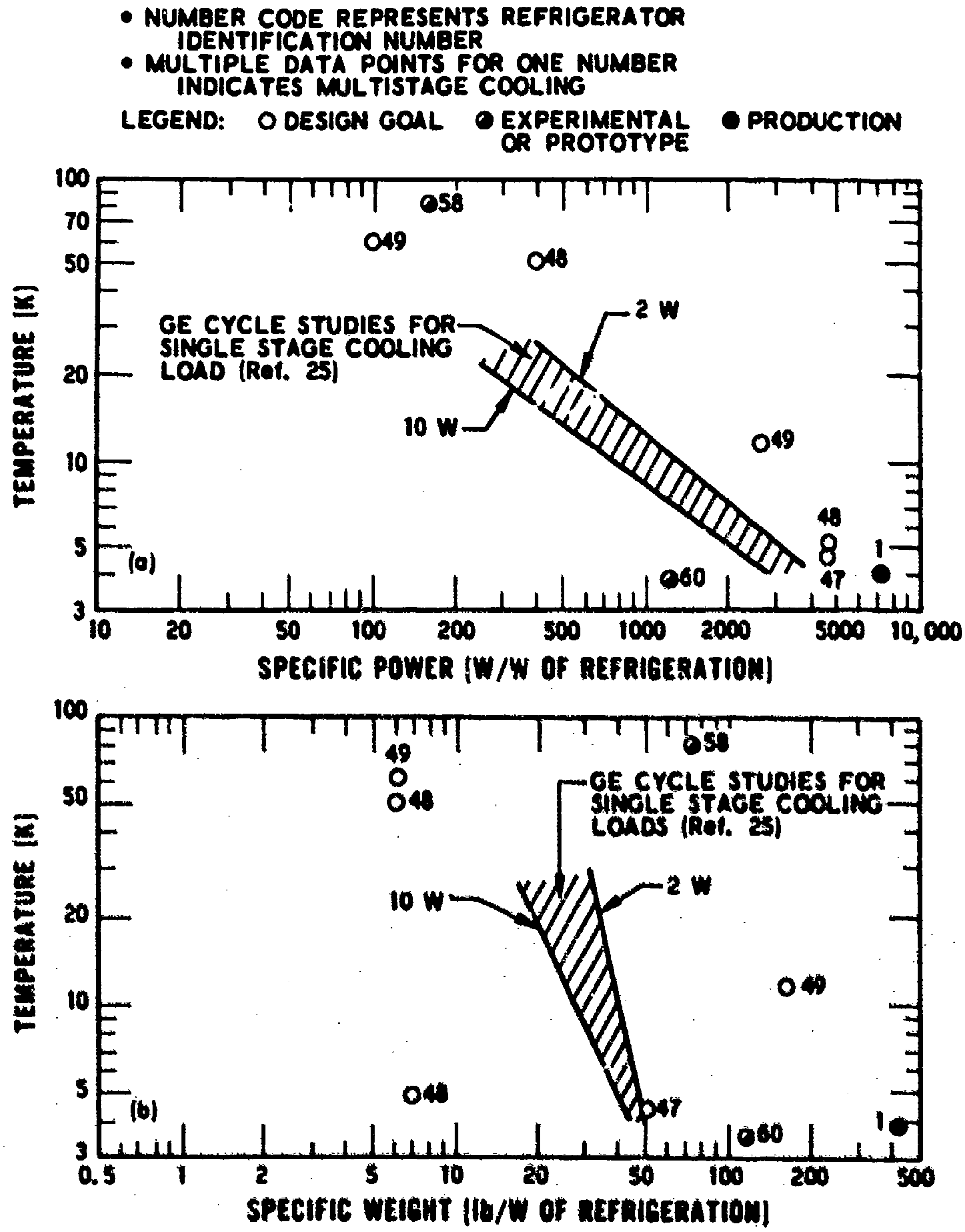


Fig. 2-12. Turbomachinery Refrigerator Power and Weight Characteristics

Table 2-9. Design Characteristics of a 2 W/4.4 K Turbomachinery Refrigerator

- Claude Cycle
- Based on General Electric Cycle Studies (Ref. 15)

Refrigerator Performance	
Load Temperature	4.4 K
Load Capacity	2.0 W
Cycle Fluid	Helium

Turboalternators	1	2
Inlet Temperature, K	75.	12.
Electrical Power Output, K	32.87	8.85
Speed, rpm	205,200.	78,900.
Overall Efficiency, Percent	37.3	36.17
Wheel Diameter, in.	0.625	0.625
Flow Rate, g/sec	0.752	1.327

Compressor	
Type	Regenerative
Number of Stages	5
Flow Rate, g/sec	2.872
Overall Pressure Ratio	2.42
Speed	105,200 rpm
Inlet Temperature	300 K
Stage Efficiency	16 Percent
Motor Efficiency	80 Percent
Heat Rejection Temperature	300 K
Power Input	1.472 kW
Compressor Weight, lb	20.4

Cryogenic Heat Exchangers

Exchanger	A	B	C	D	E
Effectiveness	0.990	0.916	0.990	0.963	0.990
Hole Diameter, cm	0.040	0.020	0.027	0.020	0.020
Low Pressure Side Pressure Drop Ratio	0.02	0.005	0.01	0.0003	0.00004
Axial Length, in.	12.00	0.51	10.8	0.47	3.08
Face Area, in.^{2}	22.80	7.40	11.43	7.02	6.98
Number of Plates	504.	42.	678.	40.	261.
Core Weight,* lb	10.65	0.17	5.28	0.15	0.99
Total Core Weight, lb			17.24		

*Does not include headers

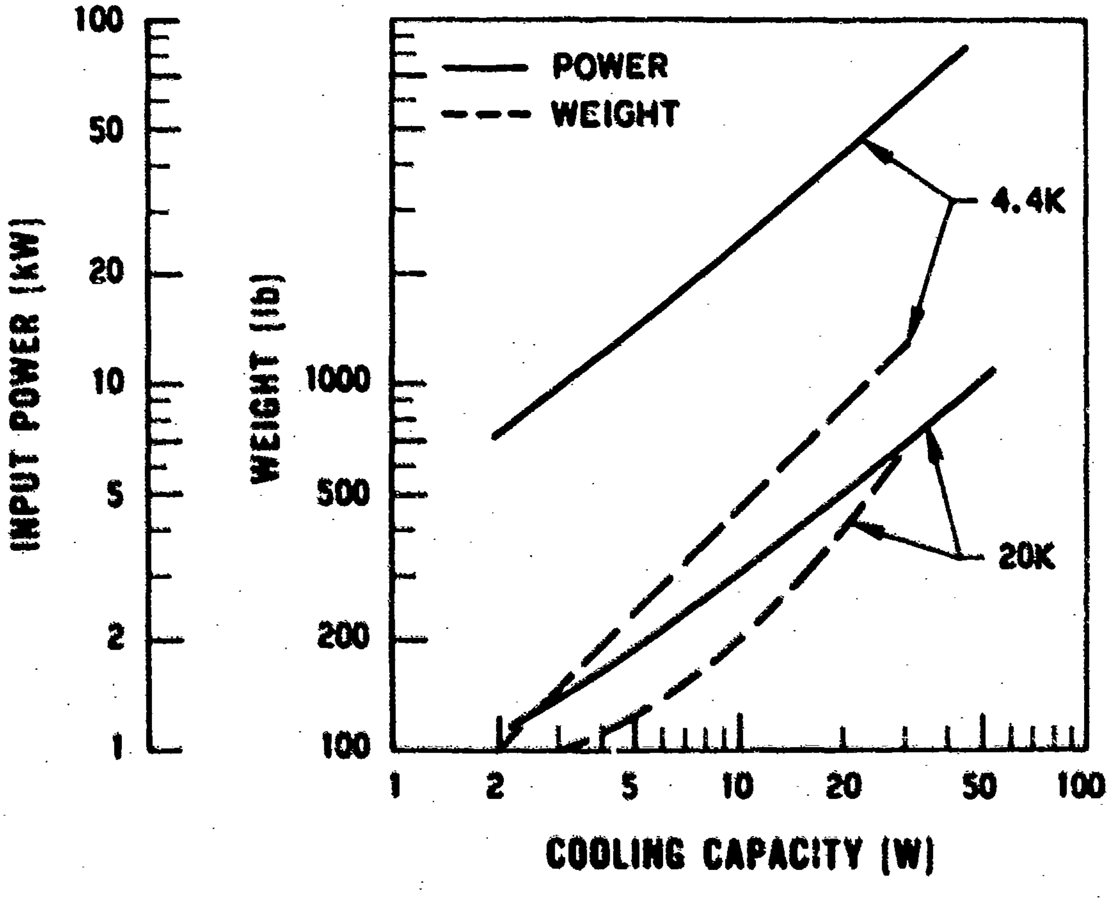


Fig. 2-13. Power and Weight Estimates for 4.4 K and 20 K Turbomachinery Refrigerators (Ref. 25)

Table 2-10. Turbomachinery Refrigerator Problem Areas (Ref. 3)

Components[a]	Problem Areas	State of Development	Number of Components
1. Counterflow Heat Exchanger	Non-metallic spacer material being used in current design to reduce longitudinal heat conduction source of contamination; fabrication of laminate layers potential source of leakage	3	5
2. Compressors	Rigid dimensional accuracy required to maintain proper clearances between rotating assemblies and thrust bearings; loosening of adjustable tilting pads	3	2
3. Turbo-Alternators	Particle contamination of the gimballed thrust bearing assembly	3	2
4. Gas Bearings	Manufacturing tolerance buildup	3	8
5. Traps/Filters	Manufacturing and improper assembly causing filter breakthrough	2	4

[a]No component appears to be life limiting.

mechanical design of the compressors and expanders. In these machines the pistons are rotated as well as reciprocated. This permits the use of ports to control gas flows and clearance seals to limit leakage. Electromagnetic actuators drive the pistons. The machinery has relatively few moving parts, all of which are completely supported on self-acting gas bearings. There are no rubbing or sliding surfaces as in conventional recriprocating equipment. The refrigeration machinery required to execute the cycle is contained in two separate units--a compressor assembly and an expander package. The system also requires radiators in spaceborne applications to reject the heat of compression plus the heat generated in the housing by electrical losses. Power conditioning equipment is required to convert the basic source of electrical power to voltages of the proper frequency, amplitude, and phase for operating the refrigerator. The concept is best illustrated by a cross-section of the compressor as shown in Fig. 2-14.

b. Development Status

Arthur D. Little, Inc. (ADL) is the sole developer and manufacturer of the RR refrigerator. During the period of 1962-1970, ADL has been funded approximately $1.3 million to study and develop various aspects of the RR refrigerator. To date, the components of one unit have been built and tested at 77 K. Another unit was designed for 3.6 K and some of the components have been fabricated.

To date, ADL developments have been directed primarily toward demonstrating feasibility of the concept. ADL is presently engaged in a systems study program on rotary free piston refrigerators. This program is funded by the Air Force Flight Dynamics Laboratory. The object of the work is to generate a computer program which can be used in system studies of the type The Aerospace Corporation has conducted for the refrigerators on the Midcourse Surveillance System program. Given inputs such as refrigeration load and temperature level, the computer program will design a rotary-free piston refrigerator to satisfy the requirements. It will tabulate

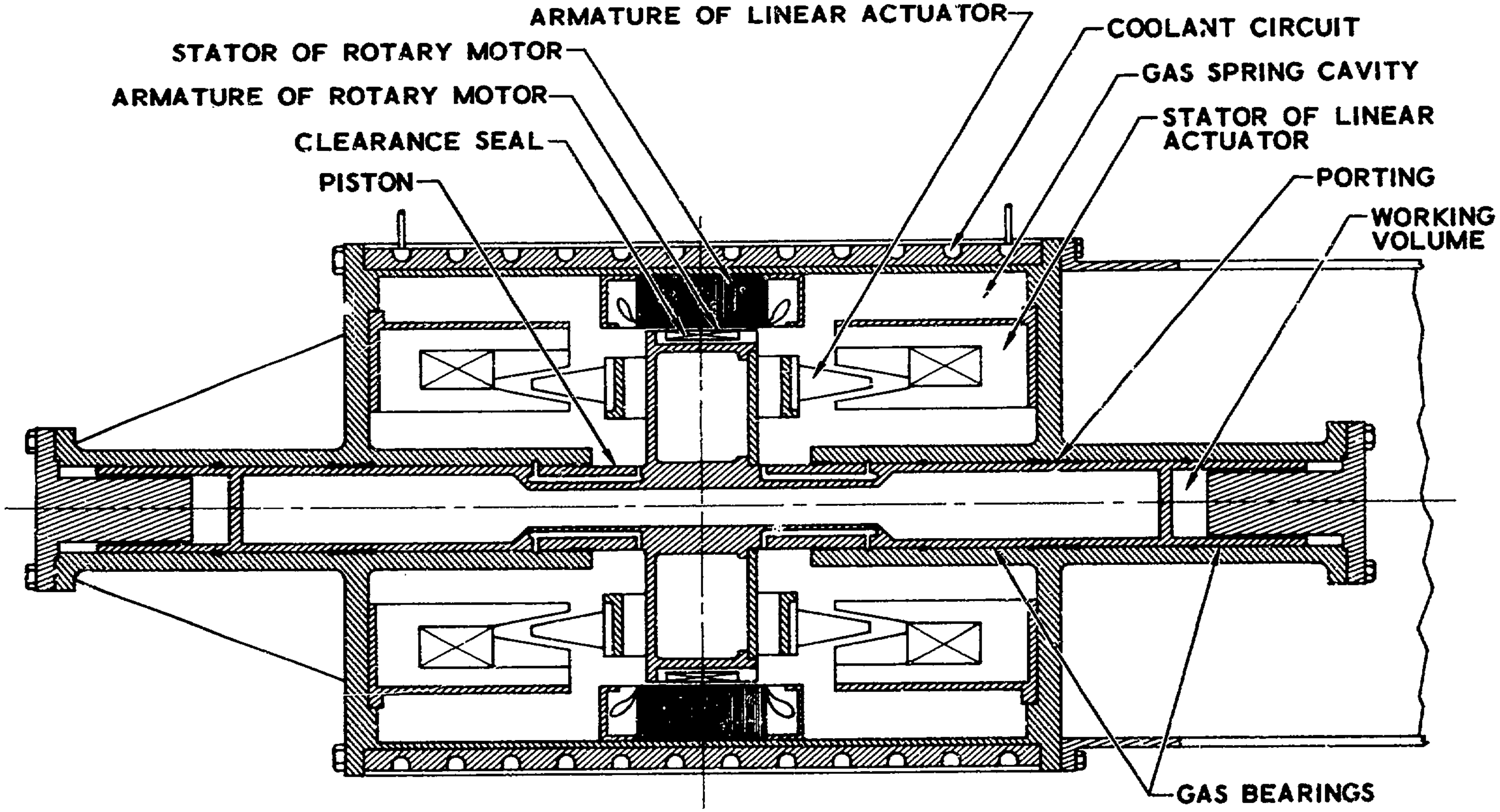


Fig. 2-14. Rotary-Reciprocating Refrigerator, Cross-Section of Compressor (Ref. 27)

the weight, size and performance characteristics of all system elements as well as the system as a whole. This system study is being undertaken on advanced development funding. It is the first step in a program which would logically lead to the design of a complete refrigeration system, and the subjecting of this system to performance tests, qualification level mechanical tests, and a 5,000-hr life test.

c. Performance Data

Since no hardware performance information is available at this time, analyses by ADL of system performance for specific applications are the primary source of data. System weight, power and volume characteristics for four RR refrigerators designed for various cooling requirements are summarized in Table 2-11. Each of the four systems as indicated provides cooling at two different levels. The specific weight of the refrigerator assembly per watt of refrigeration shown in Table 2-11 for the four systems includes only the weight of the compressor and expander assembly, and power conditioning equipment, but does not include the radiator and coolant pump weights. This was done in order to maintain the specific weight ratios consistent with data presented on other refrigeration systems in this report.

Generalized curves generated by ADL for predicting specific power requirements as a function of the cooling load and temperature are presented in Ref. 27. A summary of these curves, together with the four systems defined in Table 2-11, is presented in Fig. 2-15.

d. Development Potential

On the basis of development to date and the evaluation conducted in Refs. 3 and 24, the RR refrigerator holds promise of extended life since most of the problems associated with wear, sealing, and contamination have been essentially eliminated. However, the complexity of the RR refrigerator represents an inherent development risk since it is a relatively novel approach and only one unit has been built. A summary of potential problem areas and the state of development is shown in Table 2-12.

Table 2-11. Rotary-Reciprocating Refrigerator System Characteristics for Various Cooling Requirements (Ref. 27)

Parameter \ System Identification	(ID #63)		(ID #64)		(iD #65)		(ID #70)	
Refrigeration Load (W at K)	1.0 W at 15 K + 15.0 W at 50 K		1.0 W at 15 K + 15.0 W at 30 K		2.0 W at 20 K + 40.0 W at 60 K		1.5 W at 12 K + 40.0 W at 60 K	
Input Power (W)	1300		2160		1760		1770	
Specific Power (W/W)*	87		144		44		44	
Weights								
Compressor Assembly	88		110		100		--	
Expander Assembly	50		60		55		--	
Power Conditioning Equipment	18	156	32	202	30	185		240
Radiator	40		67		100		--	
Coolant Pump	10	50	15	114	10	140	--	---
Total (lb)		206		284		295		
Specific Weight (lb/W)**	10.4		13.4		4.6		6.0	
Sizes								
Compressor Assembly	8 in. dia. x 36 in.		8.5 in. dia. x 36 in.		8 in. dia. x 36 in.			
Expander Assembly	12 in. dia. x 48 in.		12 in. dia. x 48 in.		12 in. dia. x 40 in.			
Power Conditioning Equipment	0.7 ft^3		1.1 ft^3		0.7 ft^3			
Radiator	40 ft^2		67 ft^2		100 ft^2			
Coolant Pump	0.2 ft^3		0.3 ft^3		0.2 ft^3			

*Based on the high temperature load

**Based on weight of compressor, expander and power conditioning equipment only and the largest load of the system

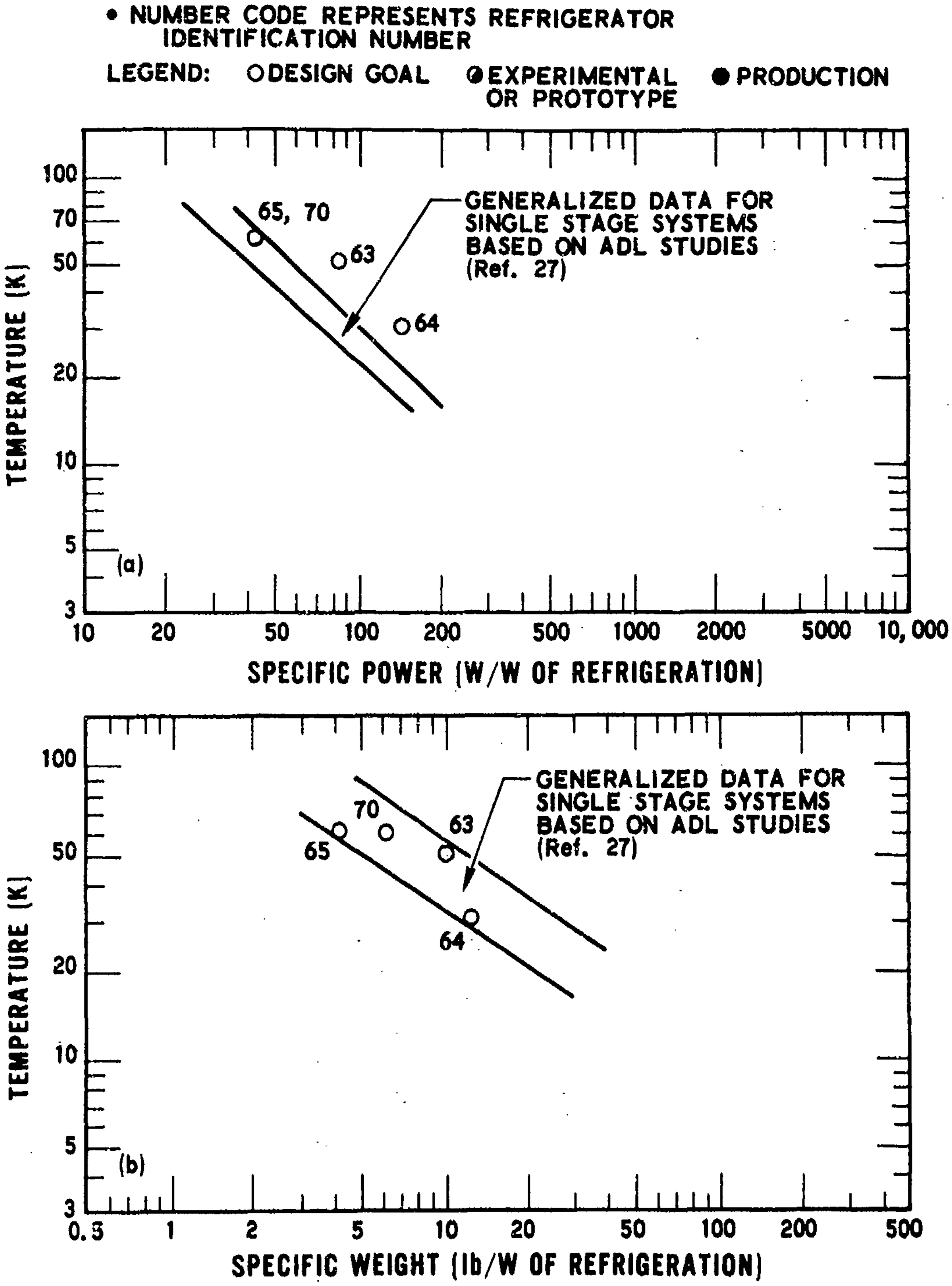


Fig. 2-15. Rotary-Reciprocating Refrigerator Power and Weight Characteristics

Table 2-12. Rotary-Reciprocating Refrigerator Potential Problem Areas (Ref. 3)

Components	Problem Areas	State of Development	Number of Components
1. Counterflow Heat Exchanger	Non-metallic spacer material being used in current design to reduce longitudinal heat conduction source of contamination; fabrication of laminate layers potential source of leakage	3	5
2. Linear Actuators*	Outgassing of the coils contaminating piston bore assemblies	4	4
3. Rotary Motors	Needs verification of performance in submerged environment	2	2
4. Gas Springs	Manufacturing tolerance buildup	3	2
5. Gas Bearings	Manufacturing tolerance buildup	2	6
6. Linkages (Reed Connectors)*	Fatigue due to load reversals and end loading	4	4
7. Traps/Filters	Filter failure due to improper assembly	2	4
8. Linear Actuator Switching	----------	---	---

*Currently life limiting, correctable by development

Status Key:

0	Fully developed	3	Currently being developed
1	Requires life test	4	Requires development
2	Performance testing	5	Beyond state of the art (1975)

G. JOULE-THOMSON (J-T) CLOSED-CYCLE REFRIGERATOR

1. BACKGROUND

Until recent years, the majority of tactical aircraft requiring cryogenic cooling utilized expendable systems consisting of either high-pressure gas bottles combined with J-T expansion or cryogenic liquids. In addition to the severe logistic and servicing problems imposed by such cooling systems, expendable coolants proved undesirable since each aircraft required service prior to each flight even if the need for the cooling system never materialized. This factor plus the high cost involved prompted the development of J-T closed-cycle mechanical refrigerators specifically designed for such aircraft requirements. Systems of this type have been utilized for infrared search and track systems in F-101, F-102, and F-106 aircraft as well as for infrared mapping in the OV-1C Mohawk aircraft. In addition, this concept is being considered for a number of advanced aircraft development programs.

2. DESCRIPTION AND OPERATION

A practical J-T refrigerator cycle is shown in Fig. 2-16. This cycle is essentially identical to the reversed Brayton cycle shown previously except for one fundamental difference. The expansion process, 4 to 5, is accomplished by expansion through a throttling valve rather than through a turbine. In the J-T system, the state point 5 lies in the two-phase region, and the heat of vaporization of the coolant is used to absorb heat from the cooling load in the process 5 to 6. In a typical application, the working fluid, gaseous nitrogen, is compressed to approximately 2500 psia in a multistage oil-lubricated recriprocating compressor. The heat of compression is removed by ram air or by a fan mounted on the compressor assembly. After compression, the gaseous nitrogen passes through an adsorber/filter component which removes oil vapor and other trace contaminants which might solidify at cryogenic temperatures. The purified, high-pressure nitrogen then enters the miniature regenerative heat exchanger, or cryostat, where it is cooled by the returning low-pressure nitrogen gas. At the end of the heat exchanger

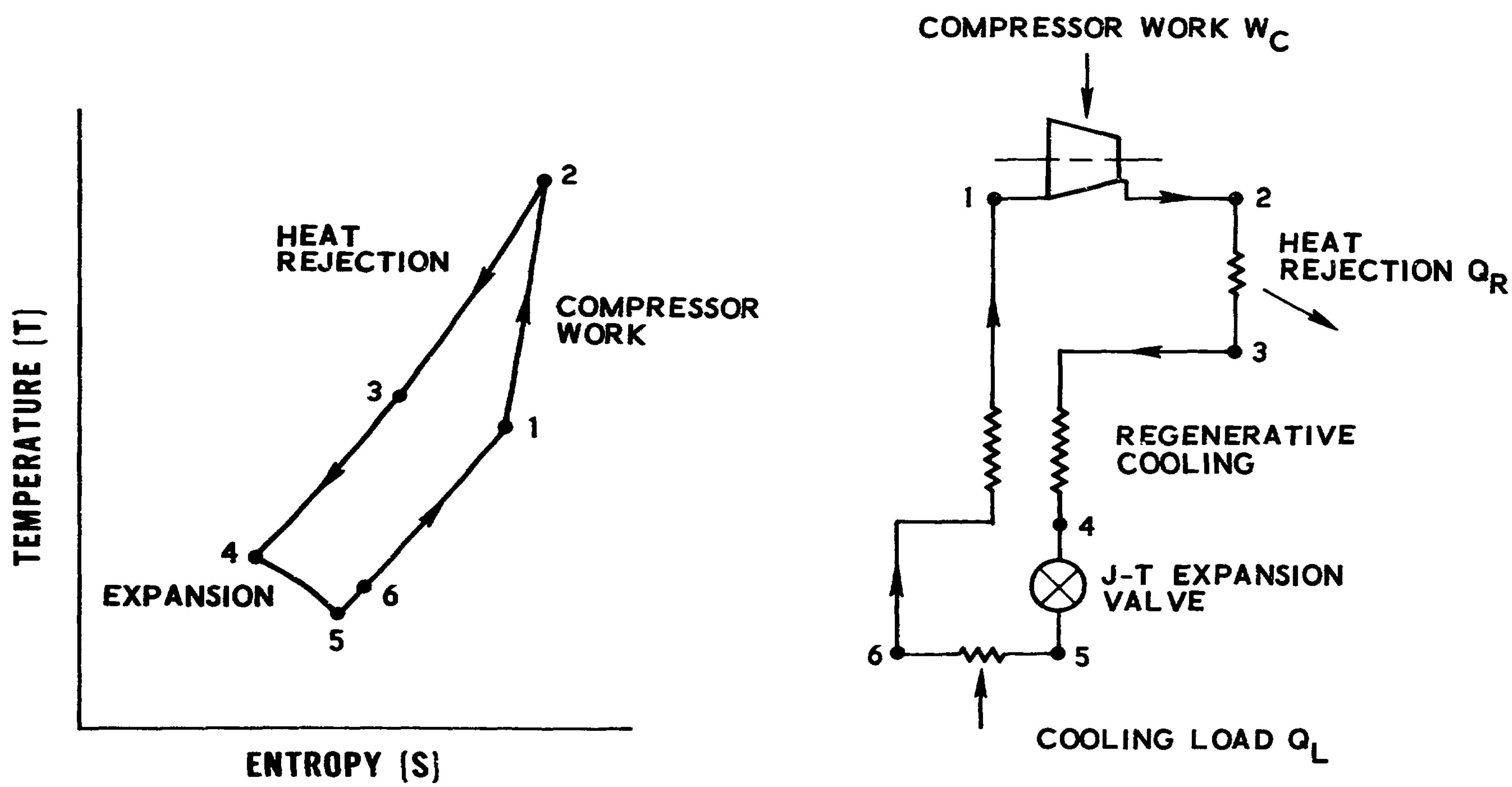


Fig. 2-16. J-T Closed-Cycle Refrigerator

the high-pressure gas is expanded, producing a temperature drop sufficient to liquefy a portion of the nitrogen. The latent heat of the liquid nitrogen is used to provide the spot cooling; the low-pressure gas, after being used for precooling the incoming gas, returns to the first stage of the compressor. A small gas reservoir (accumulator) is connected to the low-pressure return line to adjust the gas volume to compensate for the increased density of that portion of the working fluid that is liquefied in the course of normal operation.

Spot cooling is normally accomplished by conduction through a copper tip mounted on the end of the cryostat sheath. Since the cryostat is completely sealed in a sheath, the nitrogen refrigerant is at all times contained entirely within the refrigeration system. Thus the detector dewar is not subjected to system pressures, and no interface leakage problems exist.

3. MANUFACTURERS

The J-T refrigerators have been manufactured by the following companies: (a) Air Products and Chemicals, Inc., (b) Garrett Corporation (AiResearch Manufacturing Company), (c) Santa Barbara Research, (d) Hymatic Engineering Company, and (e) Hughes Aircraft Company. The majority of J-T refrigeration units produced have been open-cycle systems in which the working fluid is supplied by a high-pressure gas source. These systems which are used for short-term (hours or minutes) cooling in aircraft and some space applications are covered in Section III, Open-Cycle Systems. Based on recent information reported in Ref. 16, only three companies are presently engaged in production or development of closed-cycle J-T units: Garrett-AiResearch, Air Products, and Santa Barbara Research.

AiResearch supplies a number of closed-cycle units primarily for aircraft use (ID #15 through #20). Air Products has made two closed-cycle units (ID #55), but feels that the Solvay cycle refrigerators it has developed have much more potential for long life application. Santa Barbara Research Center produced one closed-cycle J-T unit (ID #57) for aircraft use, but is no longer active in this area.

4. PERFORMANCE DATA

Performance data are summarized in Table 2-13 and Fig. 2-17. Additional characteristics of the units are provided in Table 2-14. Specific weight versus capacity is shown in Fig. 2-18. The major advantage of J-T closed-cycle coolers is that the compressor module can be located remotely from the point of cooling and that the entire cooling system can be packaged into various configurations. Other advantages are that no adjustment is required regardless of ambient temperatures and rapid cool-down can be achieved (i.e., approximately 3 to 5 min). Service intervals between 200 and 500 hr are typical with mean time between failures of 1000 to 2000 hr.

The primary disadvantage of the J-T closed-cycle system from the standpoint of space applications is the high power requirement. For this reason it has not been extensively developed for spacecraft applications. Another disadvantage is that this system produces cooling isothermally only at the liquid temperature of the refrigerant being used which limits the flexibility of application.

H. COMPILATION OF DATA ON CLOSED-CYCLE MECHANICAL REFRIGERATORS

All the data gathered on closed-cycle mechanical refrigerators is summarized in Table 2-14 and listed in order of the refrigerator identification number. Additional details of the refrigerators which were not provided in the original identification tables are presented in Table 2-14 where available. In addition, in an attempt to provide a better comparative evaluation of the various cycles, the specific power and weight data shown for each of the cycles have been combined and shown as Fig. 2-19. It should be noted that these curves represent only gross trends in that they are estimated midpoints of the spread of data points from the individual units (i.e., Figs. 2-3, 2-8, and 2-12). More exact correlations are very difficult because of the following variations and inconsistencies in the data:

Table 2-13. Identification of Joule-Thomson Closed-Cycle Refrigerators

Identification Number	Manufacturer or Developer	Model, Description, or Program	Operating Temperature Range (K)	Typical Refrigeration Performance	Status	Power: Input (W)	Power: Input/Refrig. (W/W)	Weight: Total (lb)	Weight: Tot/Refrig (lb/W)	Remarks	Ref.
15	Garrett AiResearch	133386	77	Three cooling pts. 0.75 W ea at 77 K	Production	450	200.0	25	11.1	Utilizes ram air cooling	9
16	Garrett AiResearch	133488	77	5.0 W at 77 K	Production	650	130.0	22.5	4.5	Includes fan power	9
17	Garrett AiResearch	144406	77	3.0 W at 77 K	Production	450	150.0	19.5	6.5	Utilizes ram air cooling	9
18	Garrett AiResearch	80034	77	Two cooling pts. 1.0 W ea at 77 K	Production	460	230.0	22.5	11.25	Includes fan power	9
19	Garrett AiResearch	800398	77	Two cooling pts. 1.0 W ea at 77 K	Production	650	325.0	23.0	11.50	Includes fan power	9
20	Garrett AiResearch	800656	77	2.5 W at 77 K	Production	530	211.0	20.0	8.0	Includes fan power	9
55	Air Products	J-80-1000	~77	2.0 W at 77 K	Production	600	300.0	18.0	9.0	----------	16
56	Air Products	J-30-3500	23 and 77	2.0 W at 77 K 0.35 W at 23 K	Production	1050	300.0 1350.0	------	------	Two Stage Unit	16
57	Santa Barbara Research Center	----------	≈79	2.0 W at 77 K	Unknown	326	163.0	16.0	8.0	----------	16

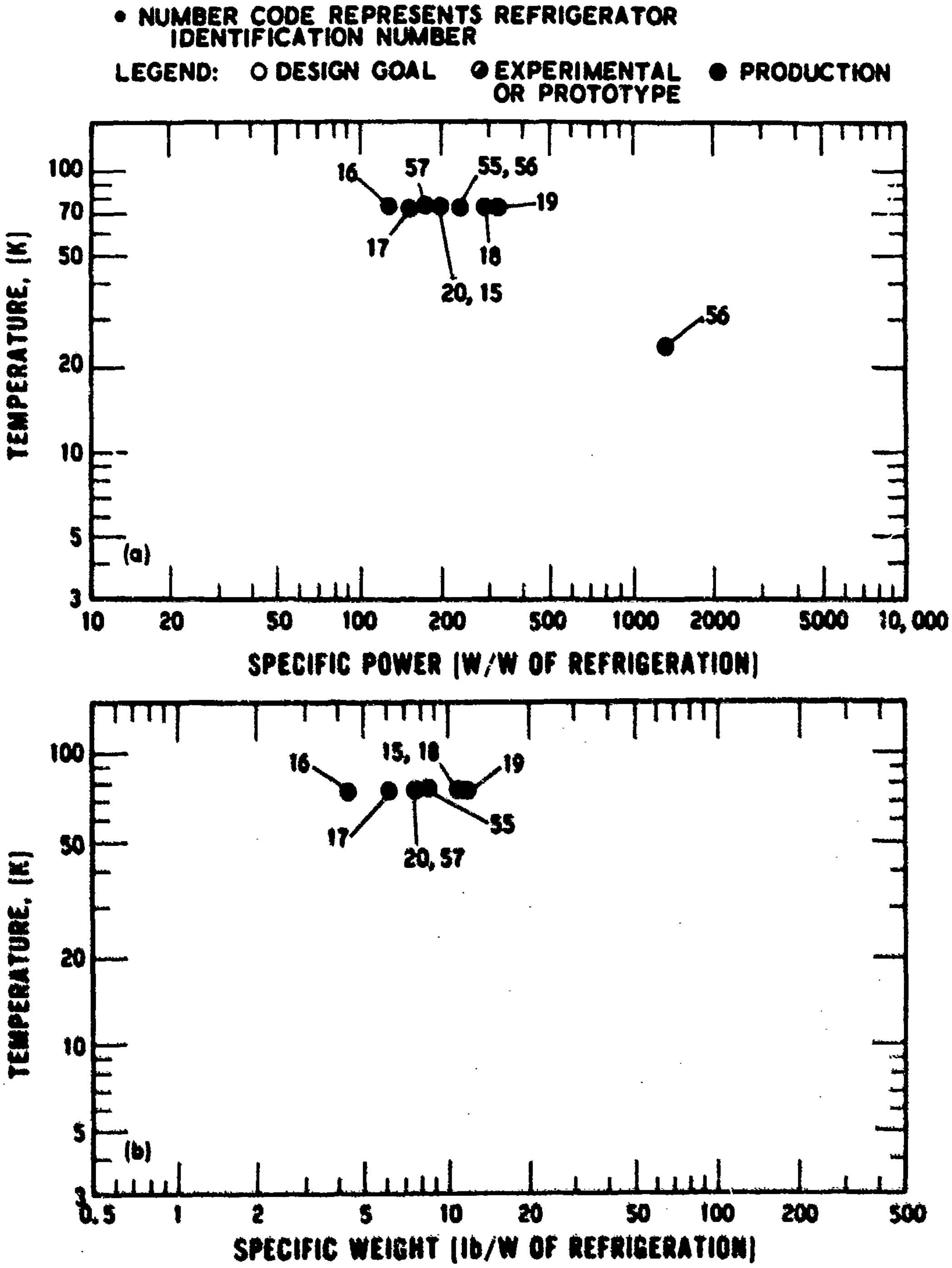


Fig. 2-17. J-T Closed-Cycle Refrigerator Power and Weight Characteristics

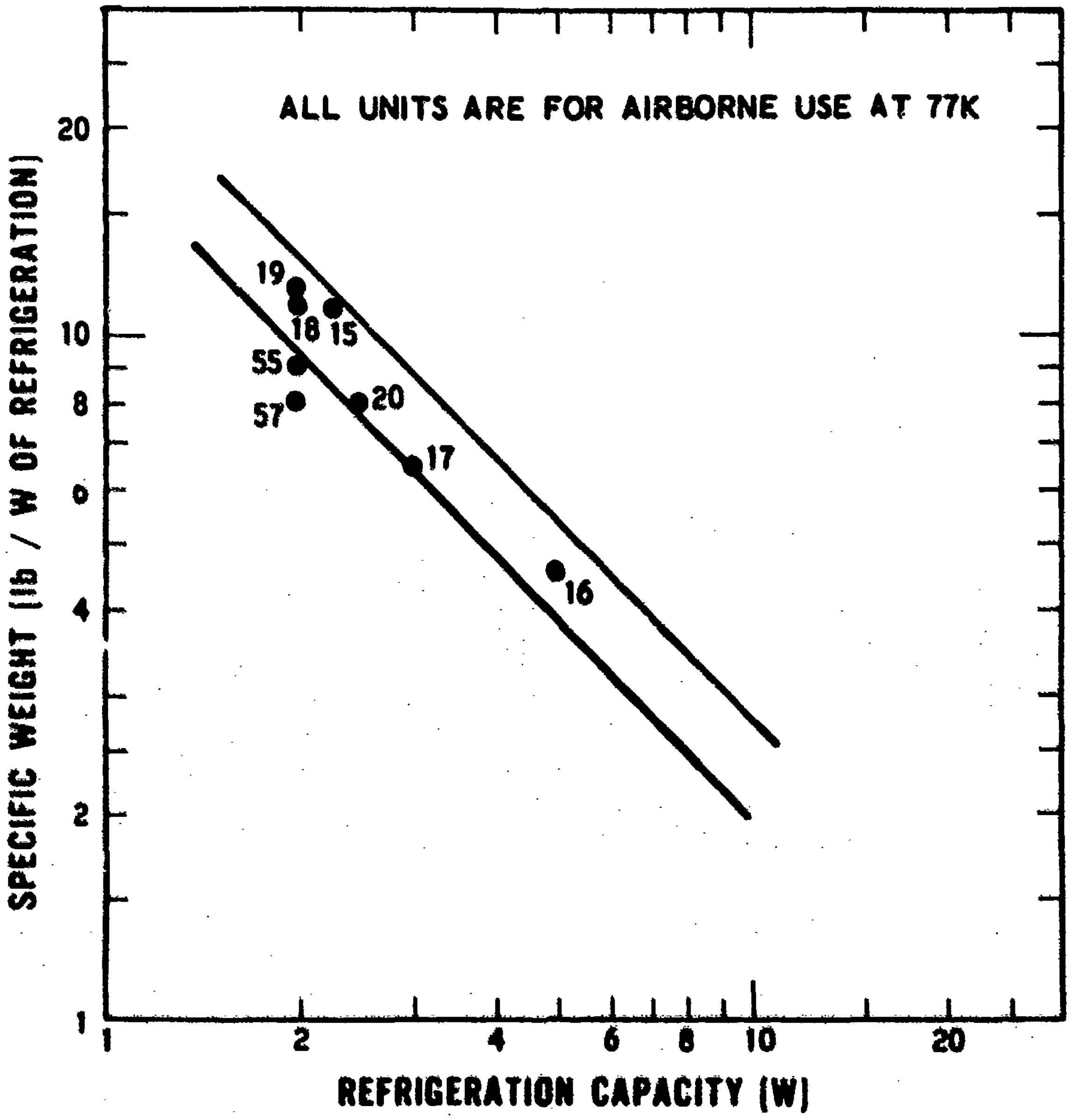


Fig. 2-18. J-T Closed-Cycle Refrigerator Specific Weight Versus Capacity

Table 2-14. Characteristics of Cryogenic Refrigerators (Listed in Order of Identification Number)

Identification # 1-10

Manufacturer	Air Products	Air Products	British Oxygen	Cryogenic Technology	Cryogenic Technology	Cryogenic Technology	Cryogenic Technology	Cryogenic Technology	Cryogenic Technology	Cryogenic Technology
Trade Name	------	------	------	Cryodyne	Cryodyne	Cryodyne	Cryodyne	Cryodyne	Cryodyne	Cryodyne
Model/Program	E-311	1-1	IR16-MK II	400	0110	1020	350	0120/RC-30	0125	077
Identification Number	1	2	3	4	5	6	7	8	9	10
Refrigeration Range (K)	3.3 to 4.5	30 to 150	12 to 23	3.8 to 4.5	6.5 to 25	12 to 28	15 to 28	19 to 28	19 to 28	25 to 150
Cycle	Claude	Solvay	Modified Taconis	Gifford-McMahon/ Joule-Thomson	Gifford-McMahon	Gifford-McMahon	Gifford-McMahon	Gifford-McMahon	Gifford-McMahon	Gifford-McMahon
Working Fluid	He	He	He	He	He	He	He	He	He	He
High Pressure (atm)	20	23	20	19.5	19	18.5	12.5	22	18.5	20
Low Pressure (atm)	0.6	10	10	1	5	5	4.5	8.5	9.5	5
Lowest Temperature (K)	3.3	30	12	3.8	6.5	12	15	19	19	25
Typical Refrigeration	1.0 W at 3.9 K	20 W at 77 K	2.5 W at 12 K	1.25 W at 4.2 K	1.0 W at 10 K	2.0 W at 13 K	2.0 W at 18.5 K	1.0 W at 26 K	1.0 W at 26 K	125 W at 77 K
Cooling Time on No Load	7 hr for 15 lb Copper	16 min	40 min	3 hr	1 hr to 12 K	30 min	45 min	15 min	15 min	30 min
Temperature Stability	±0.01 K/hr	±2 K	------	±0.12 K	±1.0 K	±1.0 K	±1.0 K	±1.5 K	±1.5 K	±2 K
Radiation Shield Temp (K)	36	None	60	90	77	------	------	------	------	None
Expander (rpm)	365	144	166	72	72	72	72	150	200	150
Power Input	7 kW	1.7 kW	2.64 kW	5.5 kW	2.1 kW	6.1 kW	2.1 kW	650 W	750W	6.1 kW
Volts-Phase Frequency	208/460-3-60	208/460-3-60	240-1-50	440-3-60	208/230-1-50/60	208/440-3-50/60	208/230-1-50/60	208-3-400	115-1-50/60	206-3-40C
Cooling Means	Air	Air	Air + Water Cooled Compr.	Air	Air	Air	Air	Air	Air	Air
Ambient Temperature Requirements	-32 C to 52 C	5 C to 43 C	-30 C	-5 C to 52 C	-32 C to 52 C	-32 C to 52 C	-32 C to 55 C	-54 C to 55 C	-32 C to 55 K	-32 C to 52 C
Required Attitude	Any	Any	Any	Compr: 10 deg horiz.	Compr: 10 deg horiz.	Compr: 10 deg horiz.	Compr: 10 deg horiz.	Compr: 10 deg horiz.	Compr: ±10 deg horiz.	Compr: ±45 deg horiz.
Cryostat Dimensions	20 x 25 x 53 in.	1.625 in. dia. 8 in. long	8 in. dia. 13 in. long	18 in. dia. 48 in. long	2.4 x 5 x 12 in.	13 x 8 x 20 in.	18 x 5 x 8 in.	------	6 x 5 x 11 in.	16.5 x 7 x 14 in.
Compressor Dimensions	27 in. dia. 42 in. long	13 x 29 x 13 in.	36 x 21 x 27 in.	30 x 32 x 54 in.	17 x 15 x 28 in.	21 x 26 x 41 in.	17 x 15 x 28 in.	------	15 x 12 x 9 in.	[illegible]7 x 15 x 15 in.
Cryostat Weight (lb)	190	10.6	20	125	30	23	22	4.5	10	28
Compressor Weight (lb)	230	130	240	800	175	425	175	21	50	115
Mean Time Between Failures (hr)	33,640	5,000 to 10,000	------	13,500	13,500	13,500	13,500	3,000	10,000	3,000
Maintenance Interval (hr)	3,000 to 6,000	5,000	------	3,000	3,000	3,000	3,000	1,000	3,000	1,000
Availability for One Unit	6 mo	3 mo	------	6 mo	4 mo	3 mo	1-2 mo	3 mo	4 mo	3 mo
Cryostat Cost/Unit	------		------	------	------	------	------	------	------	------
System Cost/Unit	$40,000 to $100,000	$5,000 to $10,000	$9,100	$75,000	$18,000	$15,500	$13,500	$8,000	[illegible]	$18,500

Table 2-14. Characteristics of Cryogenic Refrigerators (Listed in Order of Identification Number) (Continued)

Identification # 11-20

Manufacturer	Cryomech	Cryomech	Cryomech	Cryomech	Garrett AiResearch	Garrett AiResearch	Garrett AiResearch	Garrett AiResearch	Garrett AiResearch	Garrett AiResearch
Trade Name	------	------	------	------	------	------	------	------	------	------
Model/Program	GB02	GB12	AL01	AL02	133386	133488	144406	800334	800398	800656
Identification Number	11	12	13	14	15	16	17	18	19	20
Refrigeration Range (K)	7.5 to 25	9 to 30	23 to 80	23 to 89	77	77	77	77	77	77
Cycle	G-M	G-M	G-M	G-M	J-T	J-T	J-T	J-T	J-T	J-T
Working Fluid	He	He	He	He	Nitrogen	Nitrogen	Nitrogen	Nitrogen	Nitrogen	Nitrogen
High Pressure (atm)	24	24	24	24	155	155	176	110	155	176
Low Pressure (atm)	10	10	10	10	1	1	1	1	1	1
Lowest Temperature	7.5	9	23	23	75	75	75	75	75	75
Typical Refrigeration	1.3 W at [illegible].5 K	4.0 W at 13 K	1.0 W at 25 K	10.0 W at 30 K	Three cooling pts. 0.75 W ea at 77 K	5.0 W at 77 K	3.0 W at 77 K	Two cooling pts. 1.0 W ea at 77 K	Two cooling pts. 1.0 W ea at 77 K	2.5 W at 77 K
Cooling Time on No Load	25 min	35 min	12 min	25 min	6 min	12 min	6.5 min	8 min	4 min	4 min
Temperature Stability	±0.05 K	±0.05 K	±0.05 K	±0.05 K	±0.5 K	±0.5 K	±0.5 K	±0.5 K	±0.5	±0.5
Radiation Shield Temp (K)	30	30	None	None	None	None	None	None	None	None
Expander (rpm)	144	144	144	144	None	None	None	None	None	None
Power Input	3,000 W	3,000 W	900 W	3,000 W	450 W	650 W	450 W	460 W	650 W	530 W
Volts-Phase Frequency	220-1-50/60	220-1-50/60	110/220-1-50/60	220-1-50/60	115/208-3-400	115/208-3-400	115/208-3-400	115/208-3-400	115/208-3-400	115/208-3-400
Cooling Means	Air	Air	Air	Air	Ram Air	Air	Ram Air	Air	Air	Air
Ambient Temperature Requirements	------	------	------	------	-40 C to 56 C	-40 C to 75 C	-40 C to 56 C	-40 C to 56 C	-40 C to 56 C	-40 C to 56 C
Required Attitude	Any	Any	Any	Any	Any	Any	Any	Any	Any	Any
Cryostat Dimensions	5 x 5 x 21 in.	5 x 5 x 24 in.	2-1/2 x 2-1/2 x 14-1/2 in.	5 x 5 x 18 in.	0.50 ft³	1.5 ft³	0.45 ft³	1.5 ft³	1.5 ft³	1.33 ft³
Compressor Dimensions	29 x 19 x 27 in.	29 x 19 x 27 in.	------	18 x 29 x 27 in.	------	------	------	------	------	------
Cryostat Weight (lb)	25	25	5	25	25*	22.5*	19.5*	22.5*	23.0*	20.0*
Compressor Weight (lb)	175	175	------	175	------	------	------	------		
Mean Time Between Failures (hr)	5,000	5,000	5,000	5,000	1,000	1,000	2,000	------	1,000	2,000
Maintenance Interval (hr)	3,000	3,000	3,000	3,000	200	500	400	500	300	100
Availability for One Unit	6-8 wk	6-8 wk	6-8 wk	6-8 wk	4 mo	4 mo	6 mo	4 mo	4 mo	3 mo
Cryostat Cost/Unit	------	------	------	------	------	------	------	------	------	------
System Cost/Unit	$13,200	$13,200	$8,600	$10,300	$10,500	$9,000	$8,600	$12,000	$10,000	$7,000

*Total Refrigerator Weight

Table 2-14. Characteristics of Cryogenic Refrigerators (Listed in Order of Identification Number) (Continued)

Identification = 21-30

Manufacturer	Hughes	Hughes	Hughes	Hughes	Hu[illegible]	Hughes	Hughes	Hughes	Malaker	Malaker
Trade Name	------	------	------	------	------	------	------	------	Cryomite	Cryomite
Model/Program	Prototype	Flight Test Model	Prototype	------	Flight Test Model	Flight Test Model	------	------	VII-C	VII-R
Identification Number	21	22	23	24	25	26	27	28	29	30
Refrigeration Range (K)	15-75	25-75	30-75	25	70-90	75-90	80	80	25-77	40-125
Cycle	Vuilleumier	Vuilleumier	Vuilleumier	Stirling	Vuilleumier	Vuilleumier	Stirling	Stirling	Stirling	Stirling
Working Fluid	He	He	He	He		He	He	He	He	He
High Pressure (atm)	------	------	------	------	---	------	------	------	------	------
Low Pressure (atm)	------	------	------	------	------	------	------	------	------	------
Lowest Temperature (K)	15	16	------	20		------	45	35	17	49
Typical Refrigeration	0.15 W at 15 K	See Table 2-1	0.50 W at 30 K	0.80 W at 2[illegible]	[illegible] W at 85 K	1.5 W at 75 K	1.[illegible] W at 80 K	2.0 W at 77 K	2.0 W at 25 K	60 W at 77 K
Cooling Time on No Load	30 min	25 min for 100 gr Copper	30 min	15 min	[illegible]5 min	10 min	3 min	5 min	8 min	38 min
Temperature Stability	------	------	------	------	------	------	------	------	±0.02 K	------
Radiation Shield Temp (K)	------	------	------	------	------	------	None	None	------	------
Expander (rpm)	------	------	------	------	900	------	------	------	------	------
Power Input	370 W	1200 W	550 W	620 W	180 W	200 W	[illegible]00 W	580 W	480 W	1220 W
Volts-Phase Frequency	28V dc	115-3-400	28V dc	115-3-400	28V dc	28V dc	115-3-400	115-3-400	208-3-400	208-3-400
Cooling Means	Liquid	Liquid	Liquid	Air	Heat Pipe	Air	Air/Liquid	Air/Liquid	Air/Liquid	Air
Ambient Temperature Requirements	------	------	------	-32 C to [illegible]	250 F	-55 C to 71 C	-55 C to 71 C	-55 C to 71 C	mil specs	mil specs
Required Attitude	Any	Any	Any	Any	Any	Any	Any	Any	Any	Any
Cryostat Dimensions	10.5 x 13.6 x 7.8 in.	7.5 x 9.5 x 10 in.	10.5 x 13.6 x 7.8 in.	8.5 x [illegible]		6.5 x 5.7 x 5 in.	8.3 x 5.6 x 6 in.	8.3 x 5 x 6 in.	4.8 in. dia. 11.5 in. long	6.5 in. dia. 23.5 in. long
Compressor Dimensions	None	None	None	None	None	None	None	None	None	None
Cryostat Weight (lb)	------	19.5	------	1[illegible]	8.4	5.75	10	13	15.5	40*
Compressor Weight (lb)	------		------	------			------	------	------	------
Mean Time Between Failures (hr)	----	1000 (goal)	------	------	1000	5000	------	------	40,000	40,000
Maintenance Interval (hr)		1000		[illegible]00	3000		500	500	1000	1000
Availability for One Unit	------	------	------	------	------	------	------	------	Immediate	Immediate
Cryostat Cost/Unit	------	------	------	------	------	------	------	------	------	------
System Cost/Unit	------	------	------	------	------	------	------	------	$5,195	$17,[illegible]00

Total Refrigerator Weight

Total system weight including coolant loop = 4[illegible] lb

Table 2-14. Characteristics of Cryogenic Refrigerators (Listed in Order of Identification Number) (Continued)

Identification • 31-40

Manufacturer	Malaker	Malaker	Malaker	Philips	Philips	Philips	Philips	Philips	Philips (Netherlands)	Hughes
Trade Name	Cryomite	Cryomite	Cryomite	------	Prototype	Cryogem	Cryogem	Micro-Cryogem	------	Prototype
Model/Program	Mark XX	Mark XIV-A	Mark XV	------	------	42100	42151	------	N-20	Army
Identification Number	31	32	33	34	35	36	37	38	39	40
Refrigeration Range (K)	40 to 120	45 to 100	54 to 100	7 to 300	77 to 200	20 to 40	20 to 40	40 to 300	12 K	50 to 120
Cycle	Stirling	Stirling	Stirling	Stirling	Vuilleumier	Stirling	Stirling	Stirling	Stirling	Vuilleumier
Working Fluid	He	He	He	He	He	He	He	He	He	He
High Pressure (atm)	------	------	------	6	18	17	17	8.5	------	------
Low Pressure (atm)	------	------	------	3.7	28	8.5	6.5	4.5	------	------
Lowest Temperature (K)	40	45	54	7	70	20	20	40	12	50
Typical Refrigeration	110 W at 77 K	2.8 W at 77 K	1.0 W at 77 K	0.5 W at 12 K	0.50 W at 77 K	2.0 W at 30 K	2.0 W at 30 K	1.5 W at 77 K	10 W at 20 K	0.60 W at 77 K
Cooling Time on No Load (min)	7.4	7	8	15	9	10	10	5	15	20
Temperature Stability	------	------	------	------	------	------	------	------	------	±5 K
Radiation Shield Temp (K)	------	------	------	------	------	------	------	------	------	------
Expander (rpm)	------	------	------	600	750	1750	1750	1800	1750	------
Power Input	1990 W	108 W	29.5 W	700 W	70 W	350 W	550 W	90 W	1750 W	105 W
Volts-Phase Frequency	208-3-400	208-3-400	24 V dc	320-3-60	------	208-3-400	208-3-400	24 V dc	2000VA-3-50/60	------
Cooling Means	Air or Liquid	Air	Air	Water	Air	Air	Liquid	Liquid/Air	Liquid/Air	------
Ambient Temperature Requirements	mil specs	mil specs	mil specs	------	------	-54 C to 43 C	-54 C to 43 C	-55 C to 75 C	------	------
Required Attitude	Any	Any	Any	Any	Any	Any	Any	Any	Any under 0-g	------
Cryostat Dimensions	19 x 8 x 16 in.	2.9 in. dia. 13.25 in. long	2.9 in. dia. 12.25 in. long	6 x 12 x 24 in.	12 x 8 x 6 in.	6 x 5 x 11 in.	6 x 9 x 14 in.	4 x 4 x 8 in.	4 in. dia. 7.5 in. long	------
Compressor Dimensions	------	------	------	------	------	------	------	------	19 x 14 x 13 in.	------
Cryostat Weight (lb)	65^a	5.5^a	5	35	81	12	25	3	112^a	6.5^a
Compressor Weight (lb)	------	------	------	------	------	------	------	------		------
Mean Time Between Failures (hr)	40,000	40,000	40,000	------	------	1,500	1,500	------	------	3000 (est)
Maintenance Interval (hr)	1,000	1,000	1,000	------	------	600	600	500	4,000	1,000
Availability for One Unit	Immediate	Immediate	Immediate	------	------	2-4 mo	2-4 mo	Immediate	------	------
Cryostat Cost/Unit	------	------	------	------	------	------	------	------	------	------
System Cost/Unit	$24,000	$9,000	$9,000	------	------	$8,500	$8,500	$6,500	------	------

^{a} Total Refrigerator Weight

Table 2-14. Characteristics of Cryogenic Refrigerators (Listed in Order of Identification Numbers) (Continued)

Identification # 41-50

Manufacturer	Hughes	Hughes	Garrett AiResearch	Kinergetics*	Air Products	Kinergetics*	General Electric	General Electric	General Electric	Air Products
Trade Name	Prototype	Experimental	------	------	Duplex	------	------	------	------	CS-102
Model/Program	X447550	SESP - 712	ICICLE	AFFDL	Military	SRC - 07	Army ATP**	USAF ADP	USAF ADP	
Identification Number	41	42	43	44	45	46	47	48	49	50
Refrigeration Range (K)	30, 75 (Two Stage)	15, 60 (Two Stage)	75	5	5 to 300	50 to 77	4, 4	5 to 150 (Three Stage)	12 to 60 (Two Stage)	30 to 200
Cycle	Vuilleumier	Vuilleumier	Vuilleumier	Vuilleumier	Solvay	Solvay	Claude	Claude	Claude	Solvay
Working Fluid	He	He	He	He	He	------	He	He	He	He
High Pressure (atm)	27	41	54.6	------	20	------	------	------	------	------
Low Pressure (atm)	------	34	47.5	------	6	------	------	------	------	------
Lowest Temperature (K)	30	15	5	------	------	------	------	------	------	------
Typical Refrigeration	0.50 W at 30 K 6.0 W at 75K	0.15 W at 15 K 3.5 W at 55 K	0.50 W at 75 K	0.50 W at 5 K	1.5 W at 77 K	1.0 W at 38 K	2.0 W at 4.4 K	See Table 2-7	See Turbo-machinery	17 W at 77 K
Cooling Time on No Load (min)	30	30	------	------	5	------	------	------	------	20
Temperature Stability (K)	------	±0.10	------	------	------	------	------	------	------	------
Radiation Shield Temp (K)	------	------	------	------	------	------	------	------	------	------
Expander (rpm)	240	260	400	------	385	------	------	------	------	------
Power Input	480 W	540 W	Thermal (350) Elec (15)	1000 W	340 W	400 W	9.0 kW	16.0 kW	4.0 kW	1700 W
Volts-Phase Frequency	28 V dc	24-30 V dc	------	------	------	------	------	------	------	230-60
Cooling Means	Liquid	Liquid	Liquid	------	Liquid	------	------	------	------	Air
Ambient Temperature Requirements	------	------	0 to 120 F	------	Up to 65 C	------	------	------	------	------
Required Attitude	Any	Any	------	------	Any	------	------	------	------	------
Cryostat Dimensions	10.5 x 13.6 x 7.8 in.	------	Not Specified	------	1.5 x 1 x 1 in.	------	------	2 ft dia. x 4 ft long††	20 in. dia. x 65 in. long††	15 x 17 x 22 in.††
Compressor Dimensions	None	None	None	------	5 x 5-1/2 x 7-1/4 in.	------	------			
Cryostat Weight (lb)	9.6†	60†	Not Specified	------	0.50	------	------	250†	250†	------
Compressor Weight (lb)	------	------	------	------	10.5	12.4	------			150
Mean Time Between Failures (hr)	------	------	------	------	3086	------	------	------	------	------
Maintenance Interval (hr)	------	4300	2 to 5 yr goal	------		------	------	------	------	3000
Availability for One Unit	------	------	Preliminary Design	------	------	------	------	------	------	------
Cryostat Cost/Unit	------	------	------	------	------	------	------	------	------	------
System Cost/Unit	------	------	------	------	------	------	------	------	------	------

*Formerly Submarine Systems
**Advanced Technology Program
†Total Refrigerator Weight
††Total Refrigerator Dimensions

Table 2-14. Characteristics of Cryogenic Refrigerators (Listed in Order of Identification Numbers) (Continued)

Identification # 51-60

Manufacturer	Air Products	Philips	Philips	Air Products	Air Products	Air Products	Santa Barbara Research Center	Garrett AiResearch	Hymatic	A. D. Little
Trade Name	Displex	Micro-Cryogem	------	Displex	------	------	------	------	------	------
Model/Program	CS-202	P/N 460600	------	CS-1003	J-80-1000	J-30-3500	------	Prototype	Prototype	------
Identification Number	51	52	53	54	55	56	57	58	59	60
Refrigeration Range (K)	30 to 300	40 to 80	77	50 to 300	77	23 and 77 (Two Stage)	77	80	19 to 28	3.6
Cycle	Solvay	Stirling	Vuilleumier	Solvay	Joule-Thomson	Joule-Thomson	Joule-Thomson	Brayton	Brayton	Brayton
Working Fluid	He	He	He	He	Nitrogen	Nitrogen/He	Nitrogen	Nitrogen	He	He
High Pressure (atm)	------	------	30	------	------	------	------	0.72	20-30	------
Low Pressure (atm)	------	------	------	------	------	------	------	0.30	1	------
Lowest Temperature (K)	30	40	------	50	75	23	75	75	19	------
Typical Refrigeration	1.0 W at 17 K	1.0 W at 50 K	1.0 W at 77 K	1.0 W at 77 K	2.0 W at 77 K	2.0 W at 77 K 0.35 W at 23 K	2.0 W at 77 K	2.0 W at 80 K	0.3 W at 28 K	1.0 W at 3.6 K
Cooling Time on No Load	45 min	4 min	------	5 min	5 min	------	5 min	~6 hr	30 min	------
Temperature Stability	------	------	------	------	------	------	------	------		------
Radiation Shield Temp (K)	------	------	------	------	------	------	------	------		------
Expander (rpm)	------	------	600	------	3850 (compr)	3850 (compr)	------	------	1500	------
Power Input	1735 W	120 W	120 W	400 W	600 W	1050 W	326 W	375 W		1310 W
Volts-Phase Frequency	230-3-60	115-3-400	------	------	------	------	------	115-3-60		------
Cooling Means	Air	Air	Air	Air	Air	Air	Air	Water	------	------
Ambient Temperature Requirements	40 to 110 F	------	------	------	------	------	------	------	------	------
Required Attitude	------	------	------	------	------	------	------	------	------	------
Cryostat Dimensions	4 x 4 x 17 in.	1.5 in. dia. x 10.5 in. long	42 x 18 x 18 cm	1/2 in. dia. x 5 in. long	5 x 8 x 12 in.		7 in. dia. x 12.5 in. long	------		------
Compressor Dimensions	22 x 17 x 15 in.	------		21 x 15 x 11 in.		------		------	------	------
Cryostat Weight (lb)	------	4.0°	15.0°	3.3	18°	18°	16°	151*	------	124*
Compressor Weight (lb)	150	------	------	60.0	------	------	------	------	------	------
Mean Time Between Failures (hr)	------	------	------	------	------	------	------	------	------	------
Maintenance Interval (hr)	3,000	1,000	------	4,500	500	500	500	------	------	------
Availability for One Unit	Immediate	Prototype	------	Immediate	------	------	------	------	------	------
Cryostat Cost/Unit	------	------	------	------	------	------	------	------	------	------
System Cost/Unit	------	------	------	------	$9,000	------	$10,000	------	------	------

Total Refrigerator Weight

Table 2-14. Characteristics of Cryogenic Refrigerators (Listed in Order of Identification Numbers) (Continued)

Identification = 61-70

Manufacturer	Air Products	Kinergetics	A. D. Little	A. D. Little	A. D. Little	Cryogenic Technology	Cryogenic Technology	Cryogenic Technology	Hughes/Philips	A. D. Little
Trade Name	Displex	------	Rotary-Recip.	Rotary-Recip.	Rotary-Recip.	Cryodyne	Cryodyne	Cryodyne	------	------
Model/Program	MS-1003	------	Design Study	Design Study	Design Study	0120	0277	1020	Development	Design Study
Identification Number	61	62	63	64	65	66	67	68	69	70
Refrigeration Range (K)	30 to 77	77	15 to 50	15 to 30	20 to 60	19 to 30	40 to 120	13 to 20	11.5 to 75	12 to 60
Cycle	Solvay	Vuilleumier	Brayton	Brayton	Brayton	Gifford-McMahon	Gifford-McMahon	Gifford-McMahon	Vuilleumier	Brayton
Working Fluid	He	He	He	He	He	He	He	He	He	He
High Pressure (atm)	25	------	------	------	------	18.7	------	18.9		
Low Pressure (atm)	10	------	------	------	------	9.5	------	5.1		
Lowest Temperature	30	------	15	15	20	19	40	13		
Typical Refrigeration	1.0 W at 77 K	0.40 W at 77 K	1.0 W at 15 K 15 W at 50 K	1.0 W at 15 K 15 W at 30 K	2.0 W at 20 K 20 W at 60 K	1.0 W at 2[illegible] K	3.0 W at 77 K	10 W at 20 K	See Table 2-1	See Table 2-11
Cooling Time on No Load	5 min	------	------	------	------	15 min	10 min	30 min		
Temperature Stability	------	------	------	------	------	------	------	------		
Radiation Shield Temp (K)		------	------	------	------	------	------	------		
Expander (rpm)		------	------	------	------	150	333	72		
Power Input	368 W	65 W	1300 W	2160 W	1760 W	800 W	525 W	5.6 kW	2700 W	1770 W
Volts-Phase Frequency	115-3-400	------	------	------	------	208-3-400	208-3-400	220-3-60		
Cooling Means	Air	Air	------	------	------	Air	Air	Air		
Ambient Temperature Requirements	135 F max.	------	------	------	------	65 F to 131 F	-65 F to 131 F	25 F to 125 F		
Required Attitude	------	------	Any	------	------	±45 deg horiz.	±45 deg horiz.	±10 deg horiz.		
Cryostat Dimensions	1.25 in. dia. 5.75 in. long		12 in. dia. 48 in. long	12 in. dia. 48 in. long	12 in. dia. 40 in. long	1.5 in. dia. 10.35 in. long	1.0 in. dia. 5.0 in. long	------		
Compressor Dimensions	7.25 in. dia. 8.50 in. long		8 in. dia. 36 in. long	8.5 in. dia. 36 in. long	8.0 in. dia. 36 in. long	10.5 x 10.5 x 8.0 in.	10 x 9 x 6.5 in.	------		
Cryostat Weight (lb)	6.5	6	15[illegible]	202	185	5	3	33	130**	240**
Compressor Weight (lb)	14.0					20	13	425		
Mean Time Between Failures (hr)	3100	------	------	------	------	4000	------			
Maintenance Interval (hr)	1200	------	------	------	------	500	500	3000	5000	
Availability for One Unit	Immediate	------	------	------	------	------	------	------		
[illegible] Cost/Unit	------	------	------	------	------	------	------	------		
[illegible] Time	------	------	------	------	------	------	------	------		

[illegible]
**Total Refrigerator Weight

Table 2-14. Characteristics of Cryogenic Refrigerators (Listed in Order of Identification Numbers) (Concluded)

Identification # 71-80

Manufacturer	Philips	RCA	Malaker	Malaker	Malaker					
Trade Name	Prototype	------	Cryomite	Cryomite	Cryomite					
Model/Program	Army Night Vision Lab	Army Night Vision Lab	MK XVII-1	MK XVI-3	MC XV-4					
Identification Number	71	72	73	74	75	76	77	78	79	80
Refrigeration Range (K)	60 to 100	77	77	77 to 110	54 to 100					
Cycle	Vuilleumier	Vuilleumier	Stirling	Stirling	Stirling					
Working Fluid	He	He	He	He	He					
High Pressure (atm)	30		------	------	------					
Low Pressure (atm)	------		------	------	------					
Lowest Temperature (K)	60		------	------	------					
Typical Refrigeration	0.50 W at 77 K	1.6 W at 77 K	4.[illegible] W at 77 K	8.2 W at 77 K	1.0 W at 77 K					
Cooling Time on No Load	20 min	------	[illegible] min	7 min	8 min					
Temperature Stability	------	------	------	------	------					
Radiation Shield Temp (K)	------	------	------	------	------					
Expander (rpm)	------	------	------	------	------					
Power Input	90 W	Thermal	280 W	208 W	29.5 W					
Volts-Phase Frequency	------	------	208-3-400	24 V dc	24 V dc					
Cooling Means	------	Air	Liquid	Air	Free Convec.					
Ambient Temperature Requirements	------	------	To 165 F							
Required Attitude	------	------	------							
Cryostat Dimensions	------	------	3-1/4 x 14 in.	3-1/2 x 14 in.	3 x 11-1/4 in.					
Compressor Dimensions			------	------	------					
Cryostat Weight (lb)	10.3	16	13	10	5					
Compressor Weight (lb)										
Mean Time Between Failures (hr)	3000 (Estimated)	------	------							
Maintenance Interval (hr)	1000	------	------							
Availability for One Unit	------	------	------							
Cryostat Cost Unit	------	------	------							
System Cost/Unit	------	------	------	$8,500	$9,000					

Total Refrigerator Weight

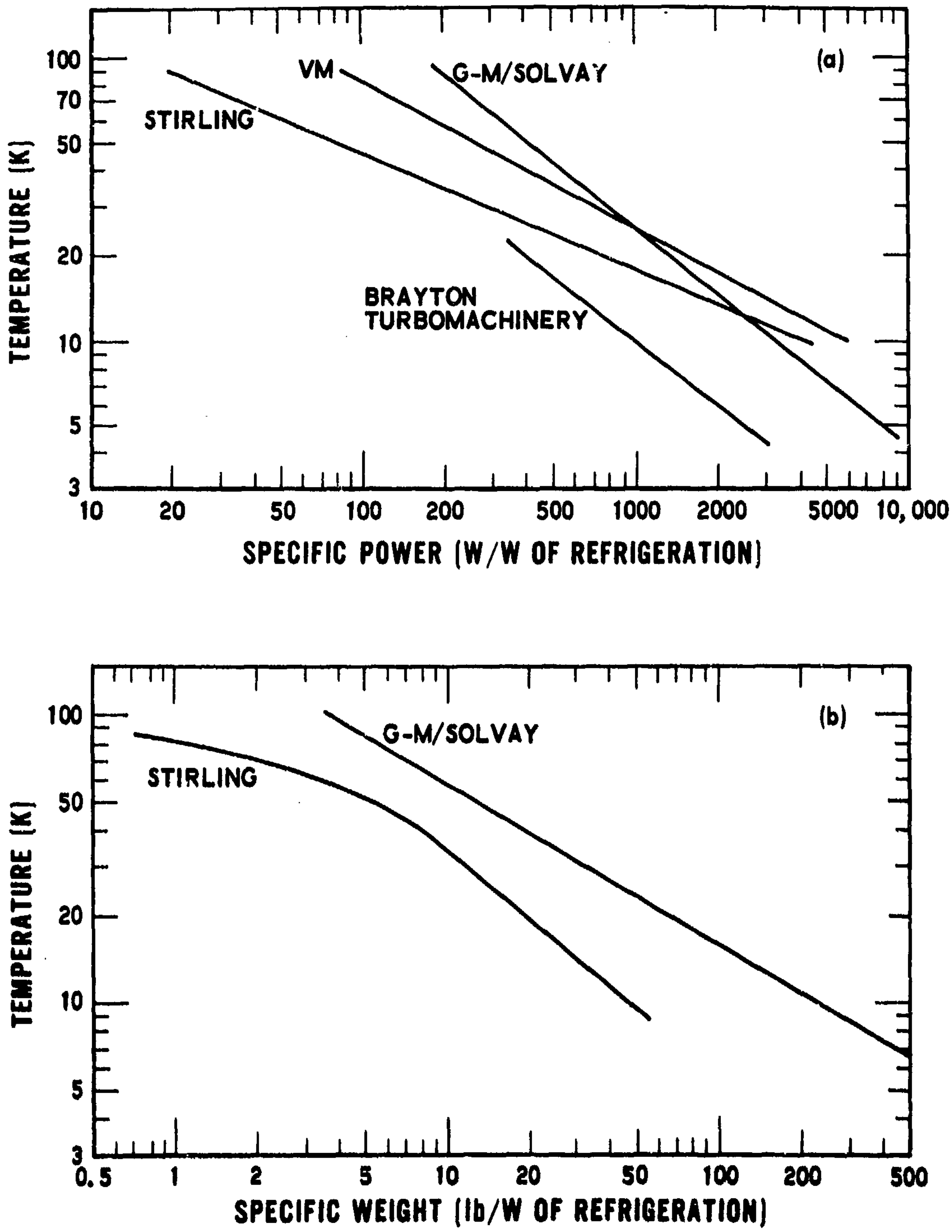


Fig. 2-19. Comparative Specific Power and Weight for Various Refrigerator Cycles

a. The data points represent refrigerator sizes ranging from a fraction of a watt to over 100 W, and obviously result in widely different specific power and weight ratios.

b. In some cases the data is based on actual production hardware, whereas in other cases the data is based on design goals or estimated weight and power based on experimental or prototype hardware.

c. The requirements and end use of each of the refrigerators cover the entire spectrum from laboratory experiments and heavy industrial use to airborne and spaceborne designed units. The obvious discrepancies in priorities of system performance, weight, power and cost provide a wide dispersion of specific power and weight.

As a result, these curves provide only a rough indication of the relative comparison of the various cycles. For the purpose of estimating the weight and power for specific applications, the individual refrigerator characteristics should be reviewed to find a unit of similar performance requirements and end useage. It should also be noted that in Fig. 2-19 specific power and specific weight curves are not shown for all the cycles described. In these cases, there was inadequate data or not sufficiently good correlations to warrant inclusion of those curves. In these cases, as previously indicated, the data provided for individual refrigerator characteristics should be reviewed to find similarity for estimating purposes.

III. OPEN-CYCLE EXPENDABLE SYSTEMS

Open-cycle or expendable refrigeration systems of interest for space application include the use of high-pressure gas combined with a J-T expansion valve, cryogenic liquids either in the subcritical or supercritical state, cryogenic solids and, for some special applications, liquids which can be stored at room temperature. The basic attractiveness of these systems is simplicity, reliability, relative economy, and negligible power requirements. In most cases, the technology is well within the present state of the art, thus requiring a minimum of development effort. The basic disadvantages of the cryogenic storage systems are the limited life due to heat leakage and the rapid increase in weight and volume for extended missions. Although high-pressure gas storage systems with J-T valves overcome the long storage limitation, the penalties associated with the storage of high-pressure gas generally make system weights prohibitive very quickly as mission duration increases.

Cooling with liquid cryogens has been used for a number of years in military aircraft applications. The logistics and costs associated with liquid cryogens, however, can become a serious drawback. Recently, many of the military systems have changed over to high-pressure gas J-T systems. In a few applications, closed-cycle J-T systems have also been used.

Included in this section are the descriptions and operating characteristics of the various open-cycle systems, weight and volume requirements, and, where applicable, commercial or military system characteristics and/or specifications.

A. HIGH-PRESSURE GAS SYSTEM

In open-cycle high-pressure gas systems, high-pressure gas (normally in the range of 1000 to 6000 psia) combined with a J-T cooler (expansion valve) produces the necessary cooling. The J-T effect is the ratio of temperature change to pressure change of an actual gas in the process of throttling or

expansion (during a constant enthalpy process) without doing work or transferring heat. Under normal pressure and temperature conditions for a perfect gas there is no cooling effect or temperature change for a throttling process. However, in actual gases under conditions of high pressure and/or low temperatures, the molecular forces become significant and cause a change in internal energy when the gas expands. The change in internal energy during the expansion process results in cooling of the gas.

It should be noted that for certain gases this effect occurs only below a specific "inversion" temperature. Helium (40 K), hydrogen (204 K) and neon (250 K), for example, require precooling to the indicated temperature before the J-T expansion effect occurs. Most other gases such as nitrogen have inversion points well above room temperature, and no special precooling is required.

1. J-T COOLERS

The J-T cryogenic cooler is based on the Joule-Thomson effect of cooling caused by the adiabatic expansion of a gas. The expanded gas, thus cooled, is passed back over the incoming gas to cool it; this results in regenerative cooling. The process continues until liquid begins to form at the orifice to produce a bath of liquid at the boiling temperature of the gas. The J-T cooler consists of a finned tube in the form of a coil, an orifice and orifice cap, and an outer shield or coil. The finned tube is made of very small inside-diameter tubing to provide the large ratio of surface area to volume necessary for effective heat exchange. Dust particles and all traces of high-freezing-point gases must be excluded from the gas entering the cooler; otherwise, the finned tube will clog or freeze up after a short period of operation. The orifice is usually surrounded by an orifice cap that returns the expanded gases back along the finned tube. This cap must be maintained in close thermal contact with the dewar or detector to ensure efficient heat transfer. The outer shield is formed in a number of ways. A separate return tube can be coiled around the finned tube, or the finned tube can be placed in a tight-fitting container so that expanded gas must flow through the channels formed by the space between adjacent turns of the finned tubing.

2. SYSTEM OPERATING CHARACTERISTICS

Commercially available open-cycle systems normally use N_2 or H_2 as the gas source and provide cooling in the range of 0.50 to 10 W at 20 to 100 K with flow rates in the order of 0.10 to 4 lb/hr. Commercially available high-purity gas with a low water content is required for operation in the cooler. To ensure that all undesirable particles are removed and that the water content is sufficiently low, a dryer and a filter can be inserted in the high-pressure line ahead of the finned tube. A chemical dryer containing an absorbent material removes the excess moisture; it is followed by a porous or a sintered metal filter to remove any floating particles. To obtain greater gas purity a cold trap consisting of a coiled tube immersed in a bath of dry ice and acetone, or liquid nitrogen, is inserted after the dryer and filter. During operation water vapor and gases, having a freezing point above that of the coolant bath, will condense on the walls of the tubing at a rate dependent on the initial purity of the gas being dried. Most systems are designed to operate over a period of a few hours or so since the weight and volume penalties for high-pressure gas storage become significant very quickly with mission duration.

3. MANUFACTURERS AND SYSTEM CHARACTERISTICS

Numerous manufacturers are involved in a number of open-cycle J-T cooling units for specific applications. Air Products probably provides the largest number of general purpose commercially available units, as well as a number of military units such as that for the Sidewinder missile. One unit (Model AC-2) which uses N_2 and H_2 was used on the Mariner program to provide 30 min of cooling at approximately 23 K.

Table 3-1 provides a representative cross-section of the various capacities and temperature ranges available. The weights and volumes of the units themselves are given and are obviously small, with the stored gas weight and volume being the primary penalty factor. The gas consumption rate is given in Table 3-1 in terms of lb/hr for the specific gas required. For a given application, the total stored gas or liquid required can be estimated

Table 3-1. Representative Data for Open Cycle Joule-Thomson Coolers

Manufacturer	Air Products	Air Products	Air Products	Air Products	Air Products	Hymatic Engineering	Hymatic Engineering	Hughes
Trade Name	Cryo-Tip	Cryo-Tip	Cryo-Tip	Cryo-Tip	Cryo-Tip	Minicooler	Minicooler	------
Model	AC-1	AC-2	AC-2L	AC-3L	AC-1-101A	MAC-215	MC-8	JTO-001
Refrigeration Range (K)	68-300	16-300	16-300	3.6-300	80-300	21, 27	80-90	87
Working Fluid	N_2	N_2 & H_2	LN_2 & H_2	LN_2, H_2 & He	N_2	H_2 & Air	Air, N_2 or O_2	Argon
Typical Refrigeration	7.0 W at 80 K	4.0 W at 24 K	6.0 W at 22 K	0.5 W at 4.4 K	0.75 W at 80 K	2.0 W at 21 K	10.0 W at 80 K	40.0 W at 87 K
Temperature Stability	± 0.10 K	± 0.10 K	± 0.10 K	± 0.10 K	± 0.10 K	± 0.10 K	± 0.10 K	------
Gas Consumption Rate at Typical Refrigeration Load (lb/hr)	3.48	3.94 (N_2) 0.25 (H_2)	0.885 K (LN_2) 0.312 (H_2)	0.885 (LN_2) 0.281 (H_2) 0.780 (He)	0.76	0.121 (H_2) 0.26 (air)	0.282 (N_2)	0.250
Refrigerator Assembly Weight (lb)	1.0	1.5	22.0	22.0	0.5	0.15	------	------
Minimum Operating Pressure (atm)	80	80	80	80	80	120	------	116
Required Attitude	None	None	30 deg of vertical	30 deg of vertical	None	None	None	None
Cool-Down Time (no load)	5 min	10 min	10 min	40 min	------	3 min	0.50 min	0.20 min
Refrigerator Dimensions	3.5 x 3.5 x 5 in.	4.5 in. dia. 7.5 in. long	6 in. dia. 17 in. long	6 in. dia. 23 in. long	------	1.0 in. dia. 2.5 in. long	0.30 in. dia. 2 in. long	------
Availability or Status	Production	Production	Production	Production	Production	Production	Production	Production
Primary Use	Laboratory	Laboratory	Laboratory	Laboratory	Laboratory	------	------	Falcon Missile
System Cost Not Including Gas Storage	$1,585	$2,[illegible]90	$3,395	$4,430	$2,850	$4,300	------	------
Reference	21	21	21	21	21	9, 77	9, 77	9

based on these numbers. The tankage penalties for the specific gas can be determined from data that is provided in Subsection III-4 below.

4. HIGH-PRESSURE GAS STORAGE

a. General Considerations

General considerations for the design and optimization of high-pressure storage vessels are fairly well-known. Basically, these relate to the desirability of minimizing container volume penalties by the use of elevated storage pressures without incurring excessive pressure shell weight. Theoretically, it can easily be shown that if the stored fluid acts as an ideal gas, the weight of the container designed to hold a given charge is substantially independent of pressure, while container volume is inversely proportional to pressure. However, in reality, gas compressibility effects are of extreme importance in pressure vessel design. At pressures over several thousands of pounds per square inch gases become less and less compressible so that volume savings at high pressures are diminished. It is therefore of extreme importance to investigate in detail the effect of initial pressure on both container weight and volume in optimization of storage vessel design.

Pressure level optimization studies for oxygen storage vessels have been conducted by Keating (Ref. 28) and others, and have indicated an optimum storage pressure of approximately 7500 psia. This pressure level was used in the design of the Project Mercury environmental control system gas supply vessels which were developed successfully and represented an advance in the state of the art of oxygen equipment.

In evaluating actual hardware weight, it must be considered that the pressure shell weight represents only 65 to 80 percent of the actual vessel weight due to various manufacturing considerations such as cut-outs, reinforcements, and welds. Thus, when vessel weights are scaled, the theoretical relations developed for optimization procedures are not necessarily valid. As a result, empirical data are also provided to aid in estimating storage penalties.

b. Design Factors

Compressibility factors have a significant influence on fluid density, especially at the higher operating pressures where the actual gas density is usually reduced from the theoretical ideal gas density. Compressibility factors for helium, hydrogen and nitrogen are shown in Figs. 3-1 through 3-3. A compressibility factor greater than one indicates a lower density than that of an ideal gas at the same conditions.

Factors of safety (i.e., design burst pressure divided by maximum operating pressure) of about 2.0 have been used traditionally in aircraft and spacecraft designs. To minimize weight and because of the extreme care taken in design and testing, Apollo pressure vessels were designed for a factor of safety of only 1.5. Materials used in the Apollo program included titanium, Ti-6 Al-4V, for nitrogen, aerozine-50, helium and nitrogen tetroxide storage; Inconel-718 for the cryogenic liquid oxygen tanks; and Ti-5 Al - 2.5 Sn for the cryogenic liquid hydrogen tanks. Low-pressure bottles (below 50 psia) for storing fluids such as water were made from 6061-T6 aluminum.

c. Weight Penalties

(1). Optimization of High-Pressure Gas Bottles

A number of studies have been made regarding optimization of high-pressure gas storage vessels. Selected data from one such study (Ref. 29) are presented in Figs. 3-4, 3-5, and 3-6. These curves show the total vessel weight per pound of useful load and the total vessel volume per pound of useful load as a function of the normal fill pressure of spherical bottles for N_2, O_2 and helium. The vessel material and safety factors are shown in the respective illustrations. It will be noted that the optimum pressure from the standpoint of weight is considerably less than that for volume. Also, since the weight curve is relatively flat, a pressure somewhat higher than that at the minimum weight point may be more appropriate where

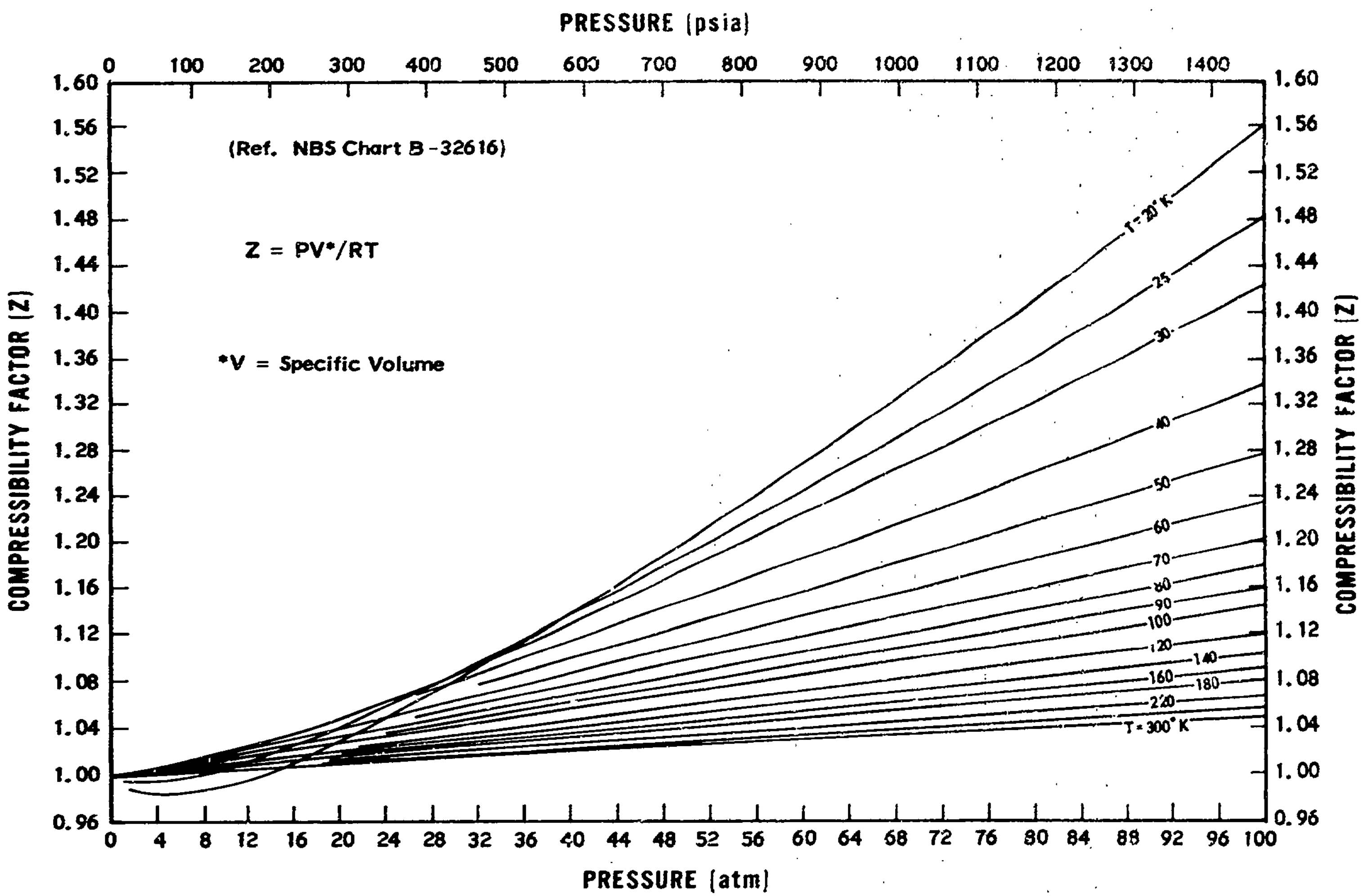


Fig. 3-1. Compressibility Factor for Helium

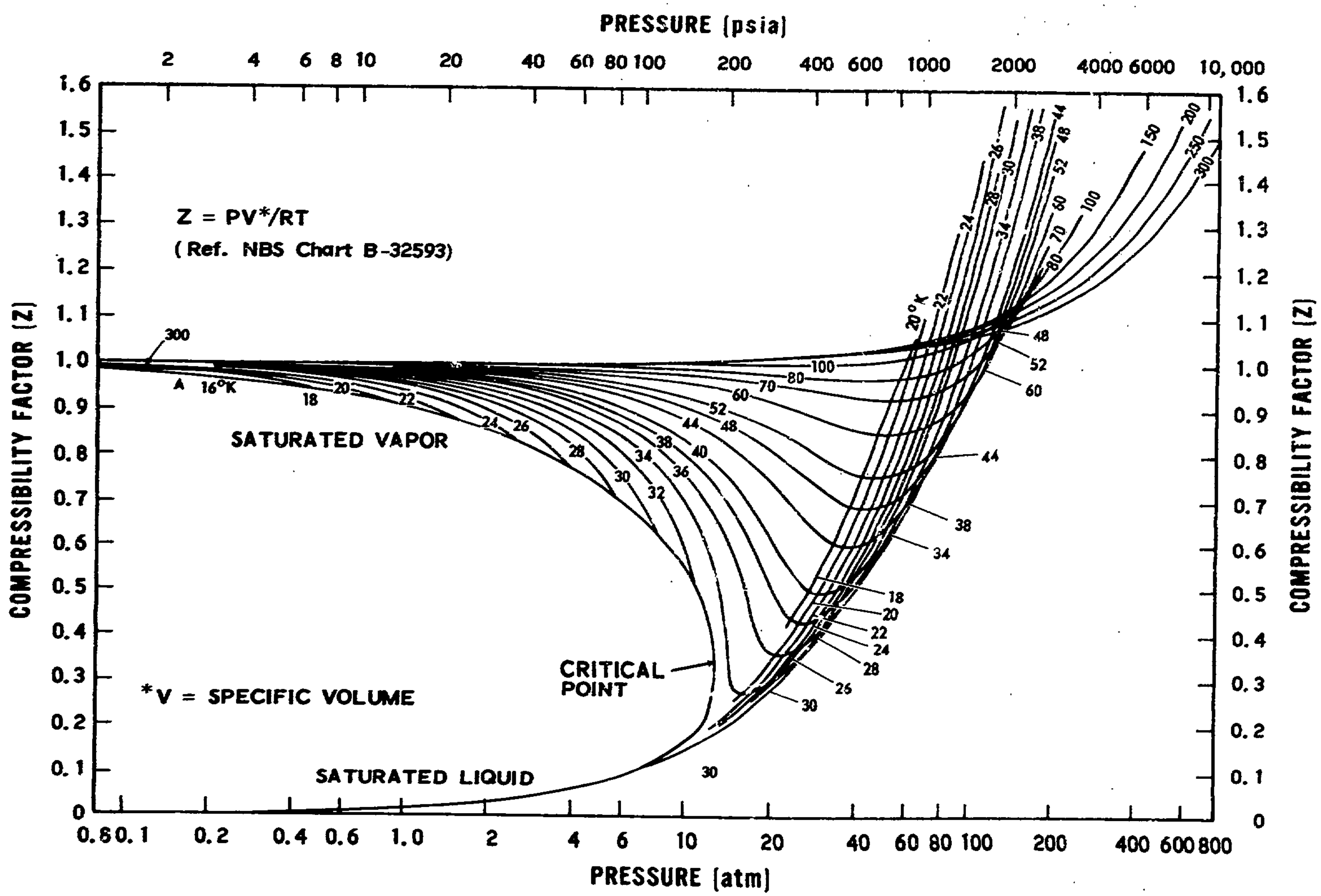


Fig. 3-2. Compressibility Factor for Hydrogen

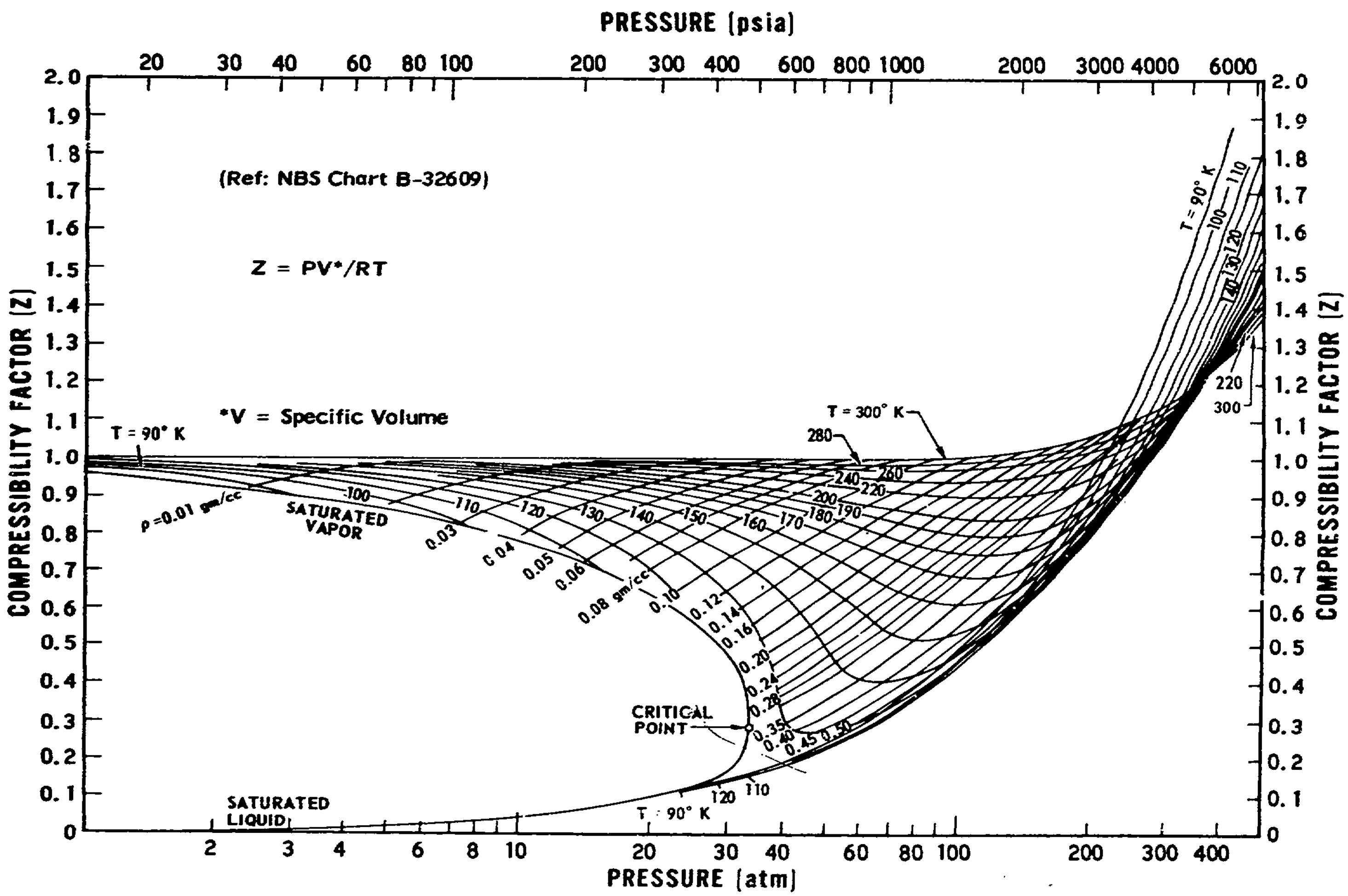


Fig. 3-3. Compressibility Factor for Nitrogen

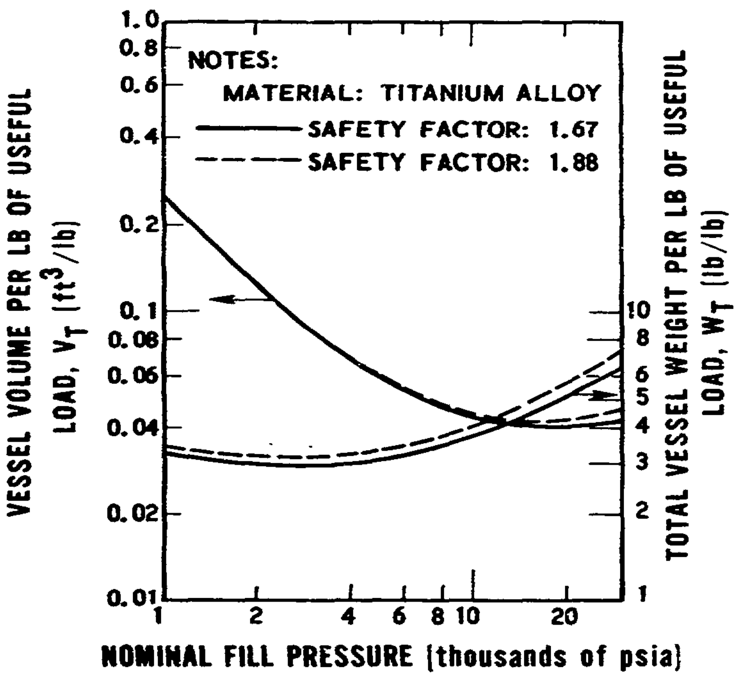


Fig. 3-4. Weight and Volume of Spherical Nitrogen Storage Vessels (Ref. 29)

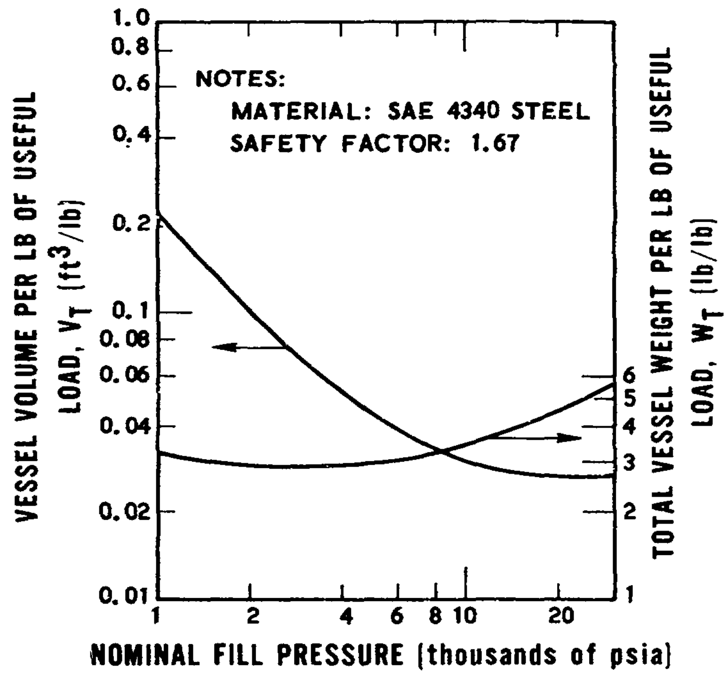


Fig. 3-5. Weight and Volume of Spherical Oxygen Storage Vessels (Ref. 29)

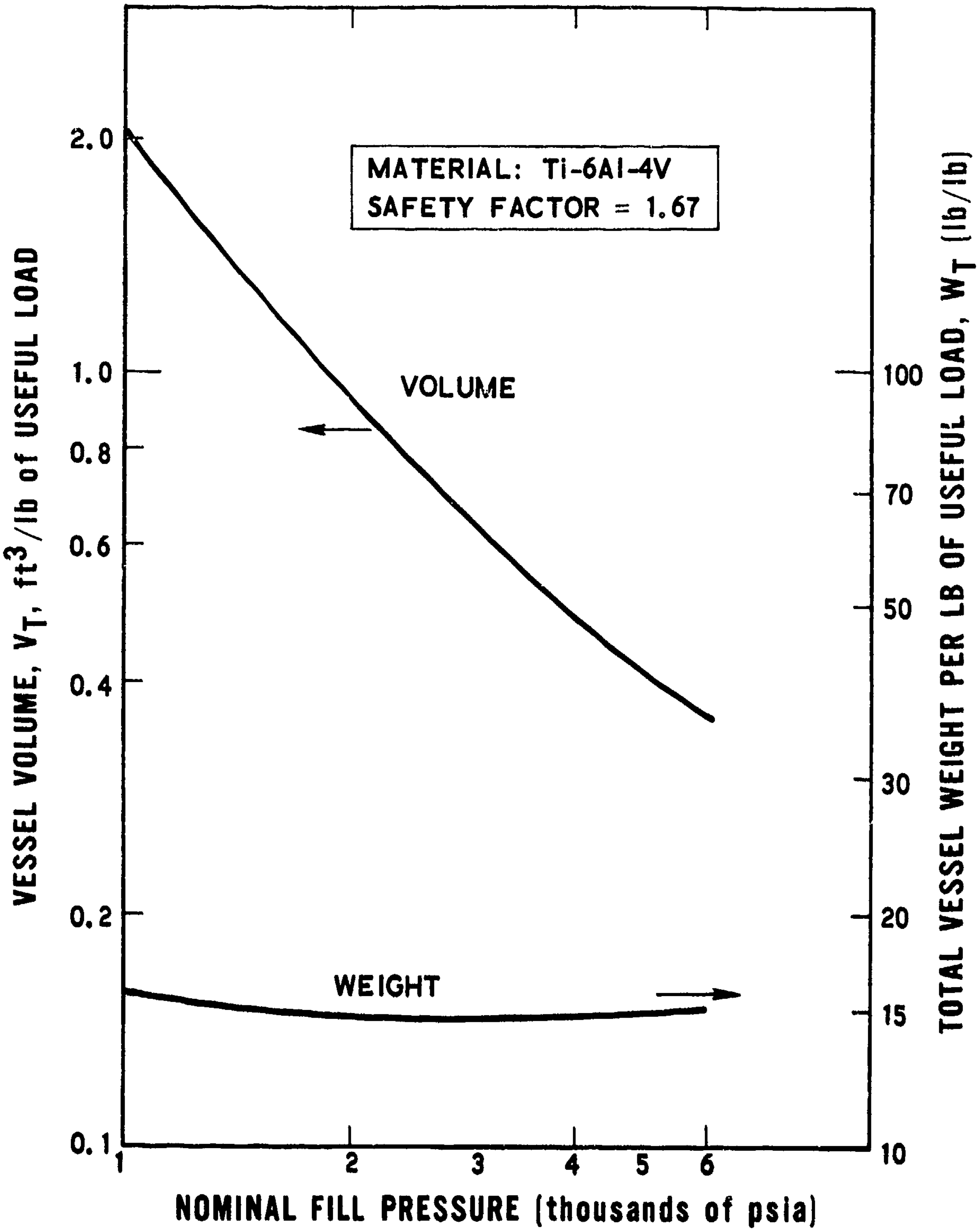


Fig. 3-6. Weight and Volume of Spherical Helium Storage Vessels (Ref. 29)

volume is also critical. Other nonoptimum factors and safety considerations will also influence the selected pressure in a practical application such as in Project Mercury where a 7500 psia oxygen bottle design pressure was utilized.

The total vessel weight is defined as dry vessel weight plus the total fluid fill weight which is based on the real density at 530 R. The fluid useful load assumes an end pressure of 30 psia. The total vessel weights do not include lines, mounting attachments, valves, etc.

(2). Empirical Data

Considerable data (Ref. 30) have been accumulated by the Mass Properties Section of The Aerospace Corporation on production cylindrical and spherical high-pressure gas bottles designed for aircraft and spacecraft systems. Curves fitted to these data, on the basis of a least squares correlation, have provided equations to predict tank weights for various design criteria. These equations allow tank weights for cylinders and spheres based on a factor of safety of 1.5 (i.e., design burst pressure divided by operating pressure) to be plotted in Figs. 3-7 and 3-8. It should be noted that these weights include only the basic tank structure and do not include associated hardware such as pressure relief and regulating valves, shutoff valves and lines, etc. It should also be noted that the equations used to product these curves apply in the range where the design pressures directly affect the tank shell thickness, and are not valid at pressures below about 2000 psia where other factors (such as minimum design gauge) may be limiting. Some specific production spherical pressure vessel data are presented in Table 3-2. This represonts selected data for titanium pressure vessels with design factors of safety between 2.0 and 2.25.

B. CRYOGENIC LIQUID STORAGE

Cryogenic fluids may be stored as liquids in equilibrium with their vapors (subcritical) or, at higher pressures and temperatures, as supercritical, homogeneous fluids. In ground or advanced aircraft operations, the fluid is usually stored in the liquid two-phase form because of weight

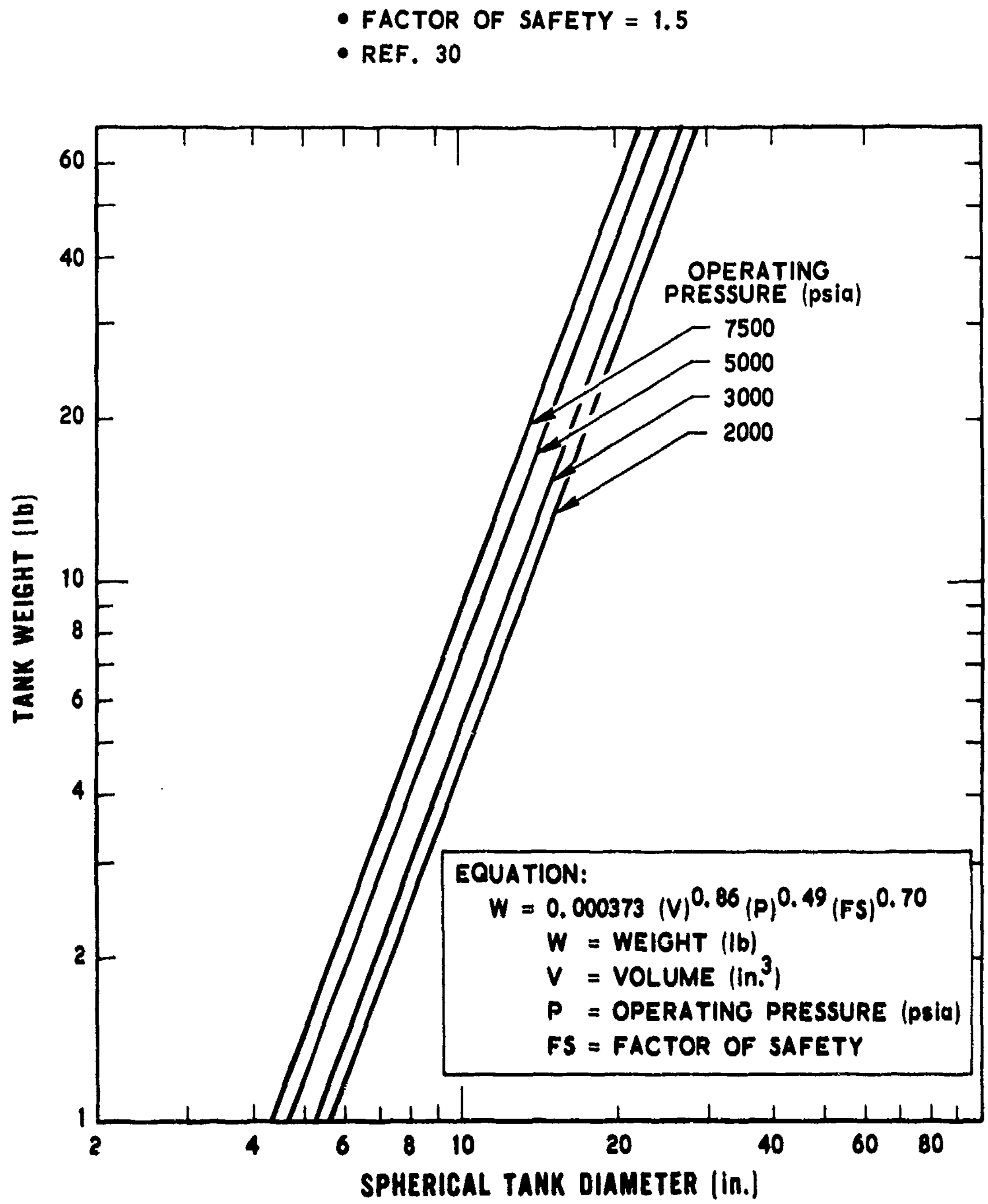


Fig. 3-7. Weight of High-Pressure Gas Spherical Tanks Based on Correlation of Production Hardware Data

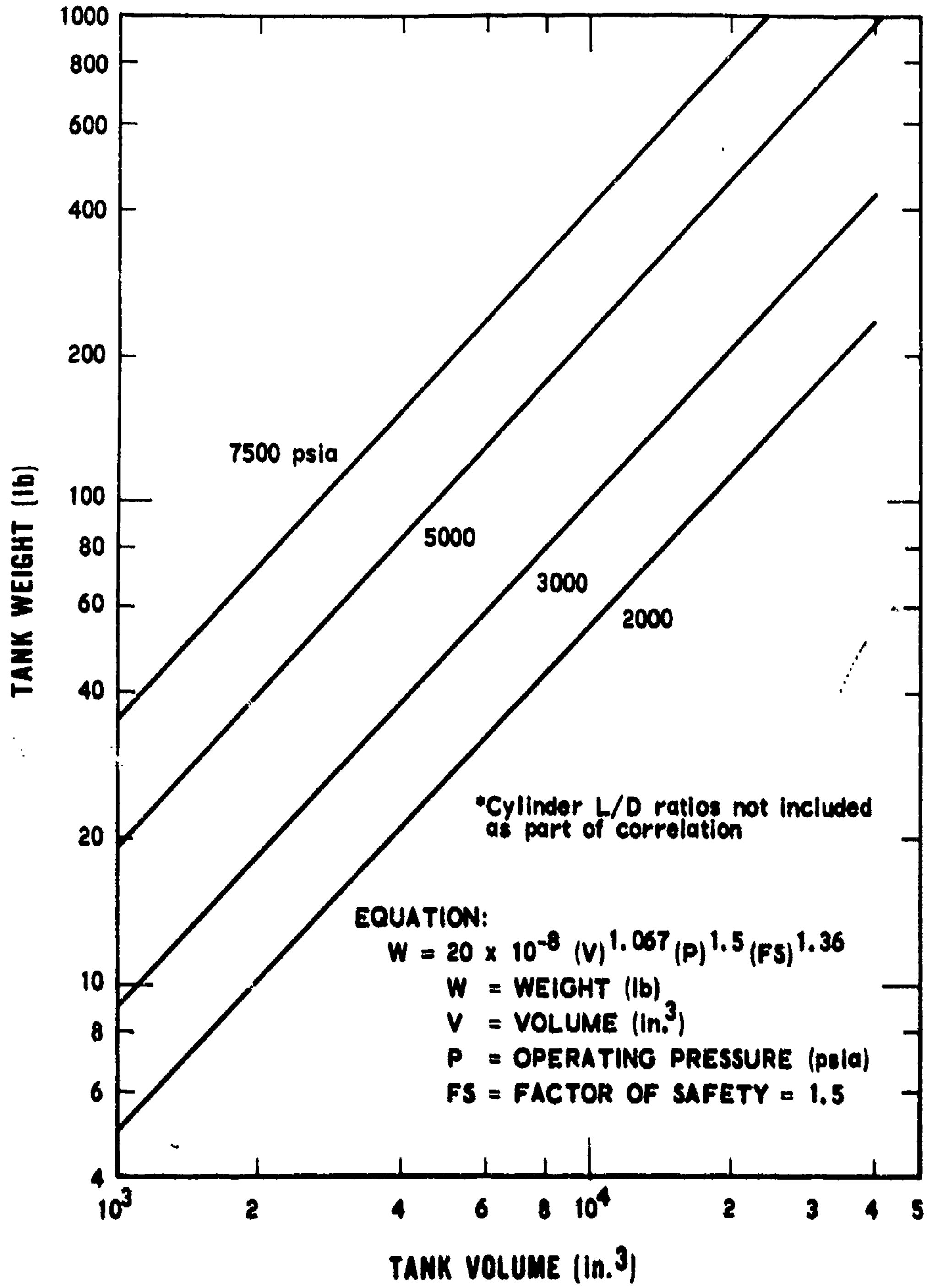


Fig. 3-8. Weight of High-Pressure Cylindrical Tanks Based on Correlation of Production Hardware (Ref. 30)

Table 3-2. Selected Data for Spherical Titanium Pressure Vessels Designed for Storage of High-Pressure Helium Gas (Refs. 31, 32)

Manufacturer	Program or Customer	Diameter (in.)	Operating Pressure (psia)	Factor of Safety	Dry Weight (lb)	Weight of Stored Helium (lb)*	Ratio of Dry Weight to Helium Weight	Ratio of Total Weight to Helium Weight**
Airtek	Agena	14.8 I.D.	3600	------	16.0	2.07	7.8	8.75
	Agena B	15.0 I.D.	3150	------	16.0	1.91	8.4	9.4
	LEM	15.5 O.D.	3000	------	22.0	1.81	12.2	13.2
	Saturn	15.4 O.D.	3000	------	21.8	1.77	12.3	13.3
Arrite	NAR	15.8 O.D.	4500	2.2	34.0	2.88	11.8	12.6
	GE	17.7 O.D.	3250	2.0	30.0	3.11	9.65	10.7
	NAR	24.5 O.D.	1000	2.2	34.8	2.83	12.3	13.3
Fansteel	Saturn IB	19.2 O.D.	3200	2.2	42.0	3.85	10.9	11.9
Menasco	NAR	16.4 O.D.	3900	2.0	44.5	2.82	15.8	16.8
	Agena	15.2 O.D.	3150	------	15.7	1.79	8.75	9.8
	Atlas	15.3 O.D.	3000	------	25.0	1.73	14.5	15.4

*Based on real gas density at 100 F
**Total Weight = Vessel plus stored helium

advantages. This subcritical pressure, two-phase system utilizes the pressurized blanket of gas maintained over the liquid surface to accomplish liquid-phase delivery. Because of gravity, the liquid is constantly exposed to the supply port of the container. The pressure differential created by the pressurized gas thus forces liquid from the vacuum-insulated storage vessel to an evaporating heat exchanger coil.

In space system storage of cryogenic liquids, the absence of gravity or acceleration orientation forces prevents the use of standard two-phase systems, since random orientation of the liquid phase during weightless conditions prevents continual communication between the liquid phase and the supply port.

Space system storage of cryogenic liquids is currently accomplished by pressurization of the cryogen to the supercritical pressure or single-phase state. The absence of gravity or acceleration orientation forces does not affect the delivery of fluid since the supply port is, at all times, in direct communication with a relatively homogeneous fluid. Although this is a satisfactory system, its weight per pound of fluid stored is greater than that encountered with the low-pressure two-phase system.

1. SUBCRITICAL STORAGE SYSTEMS

The temperatures available using baths of common liquefied gases range from approximately 4 to 77 K. By varying the pressure, the temperature of a liquefied gas can be varied to provide cooling from the triple point to the critical point. Within this range, a constant temperature cooling can be provided by pressure control. Above the critical point, temperature stability is sacrificed and the heat capacity of the liquid is smaller. (However, it will be seen later that under certain system designs the heat content in the supercritical state can be increased by using loads in series to utilize the heat content of the fluid up to a higher temperature before venting overboard.)

Selected cryogenic data for a number of fluids are shown in Table 3-3.

Table 3-3. Data for Selected Cryogenic Liquids (Refs. 33, 34, 35)

		Property								
Fluid	Symbol	Boiling Point at 1 atm (°K)	Melting Point at 1 atm (°K)	Liquid Density at B.P. (lb/ft³)	Gas Density at [illegible] (lb/ft³)	Vapor Pressure of Solid at M.P. (mmHg)	Heat of Vapor at B.P. (Btu/lb)	Heat of Fusion at M.P. (Btu/lb)	Critical Temperature (°K)	Critical Pressure (atm)
Helium	He	4.2	1 at [illegible] atm	7.8	0.0111	------	8.82	1.8	5.2	2.26
Hydrogen	H_2	20.4	14.0	4.4	0.0056	54.0	192.7	25.8	33.2	12.98
Neon	Ne	27.2	24.5	75.3	0.0562	121.0	38.3	7.2	44.5	26.8
Nitrogen	N_2	77.4	63.2	50.4	0.078	96.5	85.7	11.1	126.1	33.5
Argon	A	87.4	83.8	86.8	0.111	516.0	69.5	10.3	150.8	48.0
Oxygen	O_2	90.1	54.4	71.2	0.0892	2.0	91.6	5.9	154.1	50.1
Methane	CH_4	111.7	90.7	26.4	0.045	71.0	218.5	26.0	190.5	45.8
Ethane	C_2H_6	184.6	90.0	34.0	0.080	~0.01	210.0	40.0	305.0	48.8
Ammonia	NH_3	239.8	195.	42.6	0.048	45.0	588.0	151.0	405.0	111.2
Propane	C_3H_8	231.0	85.5	37.0	0.116	~0.01	147.0	34.4	370.0	42.0

a. Cooling Systems

Two basic types of commercially available coolers which use this concept are the direct-contact (or integral) coolers and the liquid feed coolers. The integral system is basically a detector, built right into a cryogenic dewar in which the cryogenic liquid is stored. In the liquid feed cooler, coolant is fed from a storage tank through transfer lines to a remote location. Various coolers of both types that are commercially available are summarized in Table 3-4.

(1). Integral Coolers

The integral cooler consists of a detector in direct thermal contact with a supply of liquid coolant (Fig. 3-9). The detector is integrally mounted in a dewar that serves both as the detector mount and the liquid container. When a solid coolant such as CO_2 is used, sticks of the coolant are inserted into the coolant well. Thermal contact between the solid CO_2 and the walls of the coolant well is maintained by mixing the solid with a low-freezing-point liquid such as acetone. A basic limitation of the direct-contact cooler is its operating attitude. In order to keep the coolant in direct contact with the detector, the dewar must be maintained in an essentially vertical position. For airborne and tracking instrument applications where the detector is moved through 360 deg of arc, thermal contact between the coolant and the detector is maintained regardless of the dewar attitude by using copper conducting plates that remain in contact with the coolant.

(2). Liquid Feed Coolers

The liquid feed cooler consists of an insulated liquid storage container, transfer lines, a cooling head, and the necessary controls. The transfer mechanism is either gravity or gas pressure. The gas pressure to force the liquid from the storage container to the cooling head originates from the natural pressure buildup due to thermal leakage into the storage container, or from the residual pressure of the filling operation. This basic concept is illustrated in Fig. 3-10a.

Table 3-4. Open Cycle Cryogenic Liquid Storage Systems for IR Detector Cooling (Refs. 33, 36)

Manufacturer	Model	Type: Liquid Feed	Type: Integral*	Coolant	Capacity (liters)	Operating Temperature (K)	Standby Evap. Rate (lb/hr)	Operating Time (hr)	Weight: Empty	Weight: Full	Dimensions Length x Diameter (in.)
AiResearch		X		N_2	5.0	77	0.018	30-50	7.5	16.5	10 x 11
		X		N_2	1.5	77	0.03	6	3.5	6.5	16 x 5
		X		N_2	1.0	77	0.02	3.5	4.9	6.8	7 x 6
ITT		X		N_2	1.8	77	0.04	0.3	5.9	9.3	11.5 x 6.2
LINDE	LNI-1		X	N_2	0.17	77	0.037	8.5	0.44	0.76	4 x 3
	LNI-13		X	N_2	0.28	77	0.061	8.0	4.0	4.4	5.75 x 4.5
	LNI-3		X	N_2	0.90	77		29.0	1.7	3.2	------
	LNI-4		X	N_2	0.17	77	0.031	10.0	0.36	0.67	5.5 x 4.5
	LNI-5		X	N_2	0.21	77	0.041	9.0	0.63	1.0	9 x 3
	LNI-12		X	Ne	0.78	27	0.073	8.0	4.5	5.9	8 x 4.5
	LNI-15		X	Ne/He	0.39	Ne 27	------	------	------	------	------
						He 4					
	LNI-9		X	N_2	1.8	77	1.05	------	3.5	5.6	10 x 4.5
	LNF-2	X		N_2	1.5	77	0.38	------	3.0	5.2	18 x 4.5
	LNF-4	X		N_2	3.1	77	0.026	------	7.0	12.3	------
	LNF-5	X		N_2	3.1	77	0.026	------	2.8	8.4	13.5 x 6
	LNF-6	X		N_2	0.52	77		------	0.7	1.6	12 x 3.5
	LNF-13	X		N_2	1.2	77	0.057	3.	3.5	5.5	10 x 4.0
Santa Barbara Research Center	LNI-E	X		N_2	------	77	------	8.	------	------	------
Hughes Aircraft Company	DP-999	X		N_2	1.36	77	0.03	4.	4.5	7.0	11 x 5
	DP-001	X		N_2	1.2	77	0.04	4.	7.1	9.3	------
	AP-111	X		N_2	3.17	77	0.062	8.	8.8	14.5	16 x 6

*Detector is in direct contact with coolant.

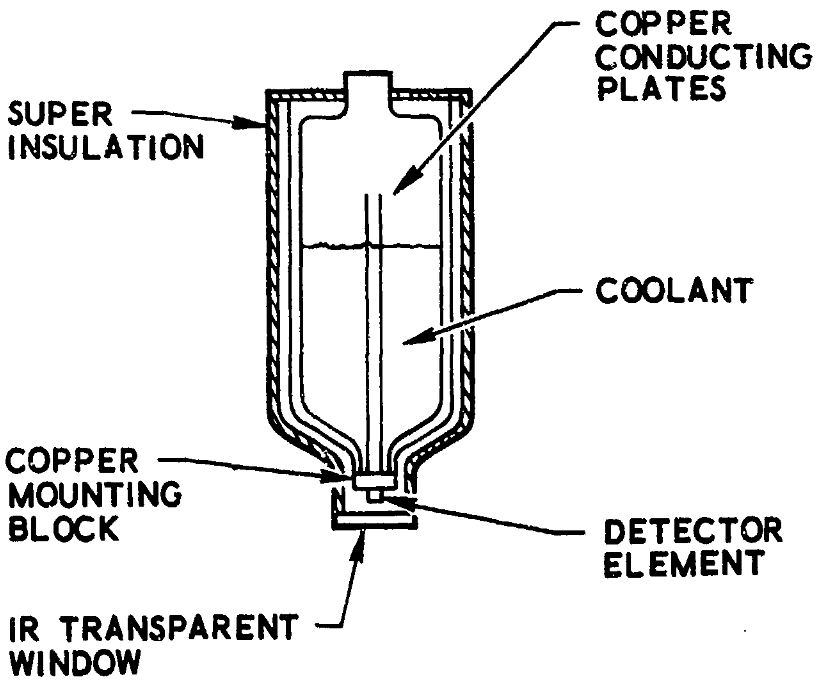


Fig. 3-9. Direct Contact Liquid Cryogen Cooler

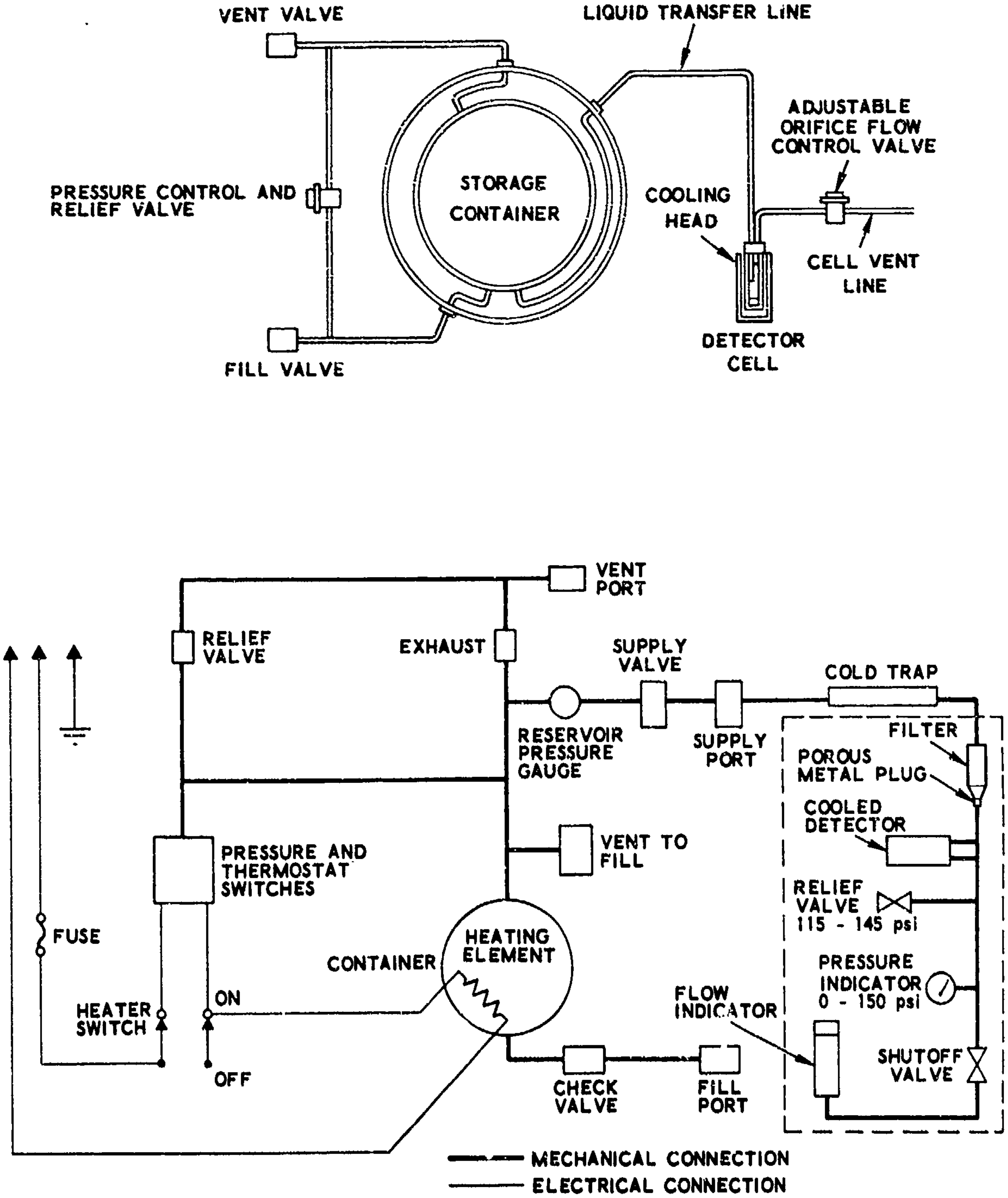


Fig. 3-10. Typical System Schematics for Liquid Cryogenic Cooled Detectors (Ref. 33)

In cases where the natural pressure buildup is not sufficient, or better regulation is required, a small pressure-regulated electrical heater is placed in the storage container to evaporate the required amount of liquid. A typical system schematic is shown in Fig. 3-10b. The flow of fluid is usually controlled to provide operation over a wide range of differential pressures, using only on-off control, and to prevent flooding of the cooling head. As the detector cell cools, the liquid in the cell evaporates and vents through the adjustable orifice-flow control valve, which sets the pressure differential between the tank and the cooling head to a valve that maintains a constant rate of flow of liquid into the cell. The pressure-control relief valve regulates the pressure exerted on the liquid, and also acts as a relief valve to vent the storage container in case of malfunction.

The usual mechanism of liquid transfer in the systems in Fig. 3-10 is a two-phase flow known as Leidenfrost transfer. When small quantities of low-temperature liquid are passed through a warm tube, some of the liquid evaporates to form a gaseous skin that keeps droplets of the liquid insulated from the walls of the tube sufficiently well so that small quantities of the liquid can be efficiently transferred. Graphs of system dry weight versus total operating time and versus required liquid capacity for a typical liquid-nitrogen detector cooling system are shown in Fig. 3-11. Based on these graphs, a detector load of 1 W operating for 80 hr would require a system weighing 6.8 lb (dry) with a liquid storage container of 4.5 liter capacity with approximately 8 lb of liquid nitrogen.

2. SUPERCRITICAL STORAGE

a. Existing Systems

The state of the art in supercritical cryogenic storage systems is typified by the oxygen and hydrogen storage systems used on the Gemini and Apollo programs. Evacuated superinsulation combined with radiation shields cooled by the vented vapor has been used to minimize the heat leak. The usual design procedure consists of providing sufficient thermal protection so

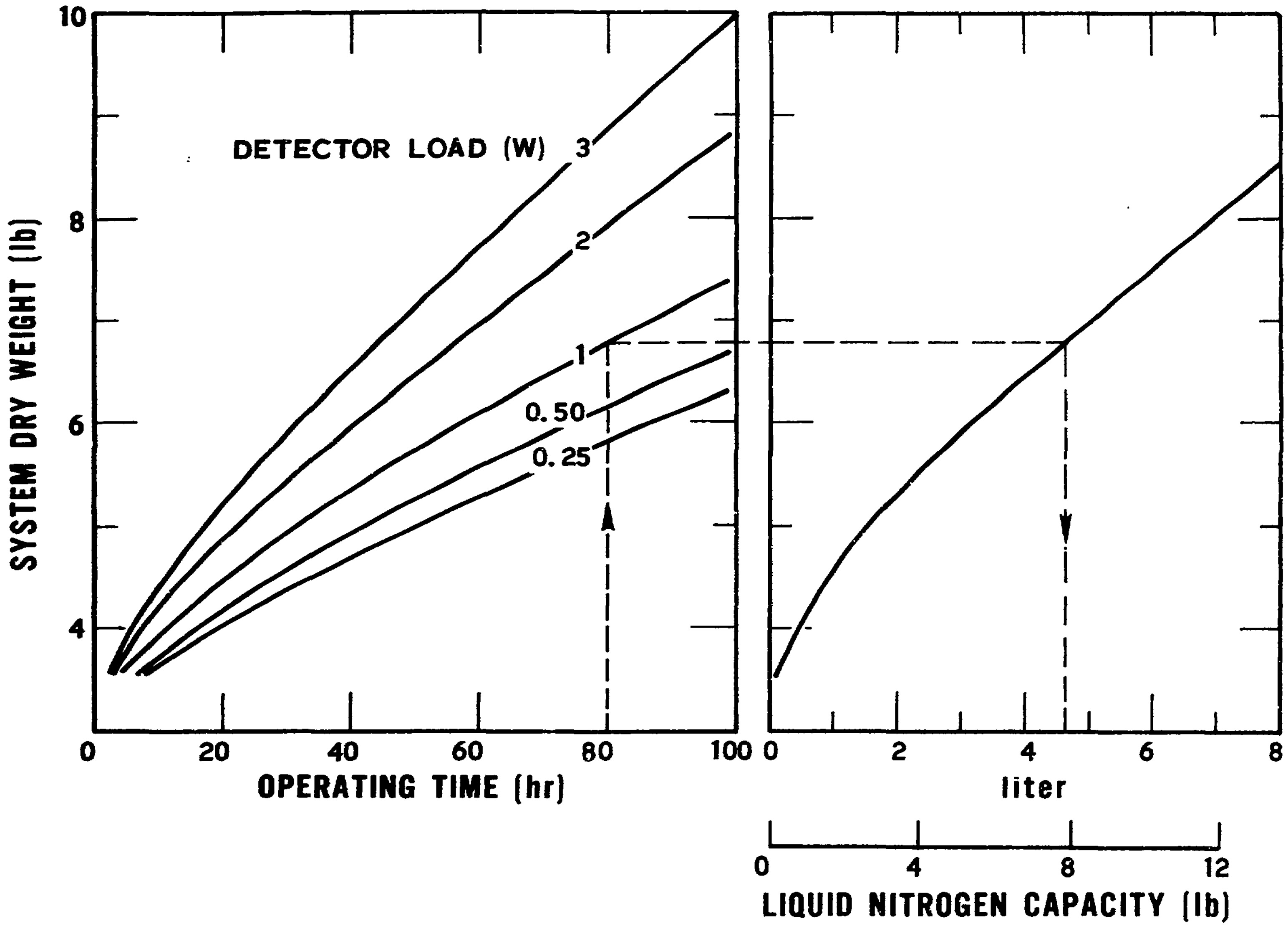


Fig. 3-11. Liquid Nitrogen IR Detector Cooling System Dry Weight and LN_2 Capacity Versus Operating Time and Detector Load (Ref. 33)

that the "boiloff" caused by heat leakage to the fluid just equals the minimum required use rate. Higher use rates are achieved by the addition of heat (by electrical resistance heating or by waste heat transfer) to the fluid or by allowing the pressure to decay during withdrawal. Normally, a nearly constant operating pressure is maintained with the use of an integral heater. These types of systems have been designed to provide gas at some higher temperature, such as at room temperature, as the ultimate product.

Characteristics of cryogenic oxygen and hydrogen tankage designed for airborne or space systems are summarized in Tables 3-5 and 3-6, respectively. Characteristics of cryogenic helium tankage designed for ground and space applications are shown in Table 3-7. The ratio of dry tankage weight to fluid weight varies considerably with size, mission length and other design requirements. The relationship of the total quantity stored to the ratio of dry tankage weight per pound of fluid is plotted for oxygen tanks in Fig. 3-12.

b. Use of Supercritical Storage for Low Temperature Cooling

If extended storage of supercritical cryogens is required to provide cooling at a specified low temperature, the design problems become somewhat different. During a constant pressure withdrawal of a supercritical fluid from a tank, the temperature of the fluid continually rises due to the added input energy required. In order to maintain a relatively constant temperature for cooling purposes, a decaying tank pressure must be utilized. Also, if the storage time requirements are such that vapor-cooled radiation shields are required, the vented fluid may not be available for shield cooling until after it has been used for load cooling because of low temperature requirements of the loads. Thus, the fluid may have to be returned from the cooling load to the tank for shield cooling before exhausting overboard. In the case of Gemini, Apollo and MOL systems, the temperature of the delivered fluid is not so critical.

Table 3-5. Summary of Typical Large Flight Weight Oxygen/Nitrogen Cryogenic Tankage

Manufacturer/Designer	AiResearch	AiResearch	Beech	Bendix	AiResearch	AiResearch	AiResearch	Essex Cryogenics	AiResearch	Beech
Program or Description	MOL	Gemini	Apollo	Development	707 Galley	727 Galley	747 Galley	C-5, C-141	Development	Development
Design Fluid	O_2	O_2	O_2	O_2	N_2	N_2	N_2	O_2	O_2	O_2, N_2, H_2
Storage State	Supercritical	Supercritical	Supercritical	Subcritical (for zero-g)	Subcritical	Subcritical	Subcritical	Subcritical	Subcritical	Subcritical
Operating Pressure (psia)	880	650	900	105	55-80	55-80	55-80	300	50-200	150
Operating Temperature (R)	160	160 (min)	160 (min)	------	------	------	------	------	------	------
Factor of Safety	1.5-2.0	------	1.5	------	------	------	------	1.5	1.5	1.5
Weights (lb)										
Dry Tankage	167.0	68.4	90.8	40.8	211	170	217	75	98	~2000
Total Fluid	715.0	180.4	330.0	25.0	250	140	350	188	184	15,730
Useable Fluid	695.0	177.4	323.0	------	------	------	------	------	------	------
Total Non-Useable	187.0	71.4	97.8	------	------	------	------	------	------	------
Total	882.0	248.0	420.8	65.8	461	310	567	263	282	17,730
Weight Ratios										
Dry Tankage/Total Fluid	0.234	0.380	0.278	1.63	0.845	1.21	0.62	0.40	0.533	0.127
Total Fluid/Total Weight	0.810	0.725	0.785	3.80	0.54	0.45	0.617	0.715	0.80	0.89
Non-Useable/Useable	0.269	0.403	0.303	------	------	------	------	------	------	------
Dimensions (in.)	(Sphere)	(Sphere)	(Sphere)		(Cylinder)	(Cylinder)	(Cylinder)	(Sphere)	(Sphere)	(Sphere)
Inside Diameter	32.5	20.56	25.1	25 x 18 x 16	L = 43.2	L = 23.9	L = 54.5		20.7	91.0
Outside Diameter	34.3	22.94	26.5	Overall	D = 23.6	D = 16.9	D = 20.5	D = 24	25.7	109.0
Flow Rates (lb/hr)										
Normal	0.90	------	0.80	2-5 Liter/min				~60	0.136	------
Minimum	0.60	0.49	0.25					(400 Liter/min)	0.093	0.152
Maximum	1.92	2.2	10.4						1.2	------
Internal Heater Power (W)										
Normal	100.0	------	417.5		None	None	None	None	None	------
Maximum	300.0	78.0	835.							
Heat Leakage										
Total (Btu/hr)	24.75	15.2	27.2	5.7					11.4	13.1
Per Unit Area (Btu/hr-ft^2)	0.98	1.33	2.04						1.17	0.066
Tank Status	Development	Production	Production	Prototype	Operational	Operational	Operational	Operational	Qualified	Thermal test article
Reference	37	38	39	40	41	41	41	42	37	

Estimated
Based on test data utilizing nitrogen

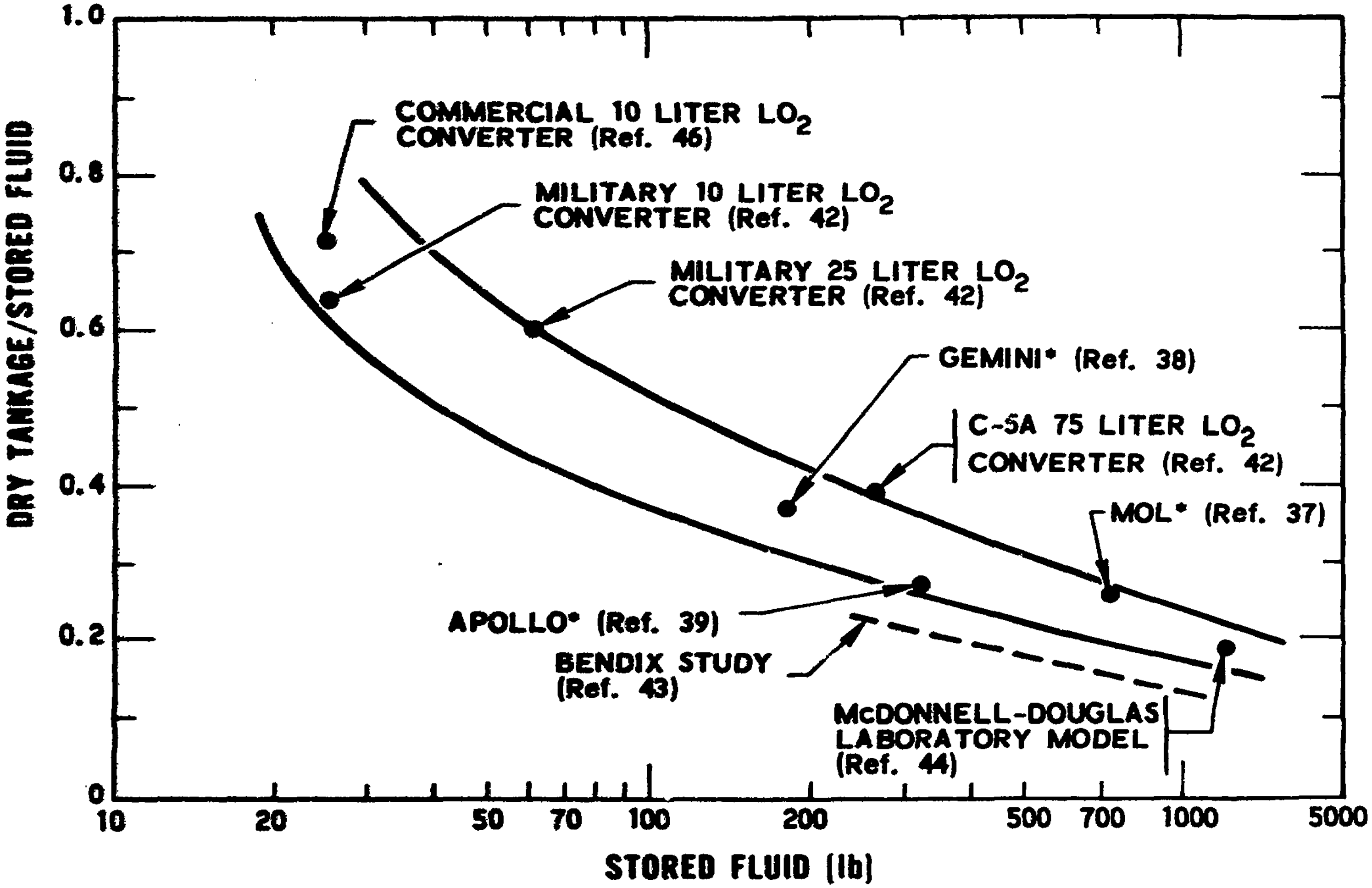
Table 3-6. Summary of Typical Large Flight Weight Cryogenic Hydrogen Tankage

Manufacturer/Designer	AiResearch	AiResearch	Beech	Bendix	McDonnell-Douglas	GE/McDAC	AiResearch	Bendix
Program or Description	MOL	Gemini	Apollo	Proposal (1 Yr. Tankage)	Development Study	Experimental	Development	AAP Development
Storage State	Supercritical	Supercritical	Supercritical	Supercritical	Supercritical	Subcritical	Subcritical	Supercritical
Operating Pressure (psia)	280	250	245	1028	1000	15	100	290
Operating Temperature (R)	37-200	37-140	37-260	------	------	38	------	------
Factor of Safety	1.5-2.0	------	1.5	------	2.2	------	------	------
Weight (lb)								
Dry Tankage	126.2	47.3	80.0	1325.2	247.0		53.1	92
Total Fluid	83.6	22.9	29.3	258.0	75.0		23.5	29
Useable Fluid	79.0	21.9	28.2	233.0	72.75		22.5	------
Total Non-Useable	131.2	47.3	81.1	1350.0	249.25		54.1	------
Total Wet Weight	210.2	69.2	109.3	1583.0	322.0		76.6	121
Weight Ratios								
Dry Tankage/Total Fluid	1.51	2.06	2.76	5.15	3.3		2.28	3.2
Total Fluid/Total	0.398	0.331	0.269	0.162	0.232		0.294	0.24
Non-Useable/Useable	1.66	2.16	2.87	5.80	3.41		2.40	------
Dimensions (in.)								
Inside Diameter	40.25	26.24	28.2	60.0	39.0	~92	26.1	28
Outside Diameter	42.6	28.90	31.8	------	------	~96	31.1	32.4
Flow Rates (lb/hr)								
Normal	0.10	------	0.0725	------	------		0.0172	------
Minimum	0.0745	0.07	------	------	------		0.0116	------
Maximum	0.225	0.28	1.02	------	------		0.07	------
Internal Heaters (W)								
Normal	35.0	------	10.0		400.0		------	------
Maximum	70.0	18.0	20.0				------	------
Heat Leakage								
Total (Btu/hr)	6.75	5.4	9.0	------	17.0	15-17	2.98	2.9
Per Unit Area (Btu/hr-ft^2)	0.19	0.360	0.291	------		0.075-0.0935	0.192	0.126
Status	Development Tanks	Production	Production	Analytical	Development	Ground Test	Development	Development
Reference	37	38	39	43	44	45	37	------

Table 3-7. Summary of Typical Cryogenic Helium Tankage

Manufacturer/Designer	North-American Rockwell	Garrett-AiResearch	Beech Aircraft	Minn. Valley Engineering	Cryogenic Engineering Company	A. D. Little	Garrett-AiResearch
Program or Description	Celestial IR Mapping (CIRM)	Apollo LEM	Apollo LEM Ground Service Tank	Commercial 1000 L Dewar	Commercial 500 L Dewar	Trailbrazer Rocket	HEAO* (NASA)
Storage State	Supercritical	Supercritical	Subcritical	Subcritical	Subcritical	Subcritical	Subcritical
Pressure (psia)							
Operating	55-38 psia	1300	------	15.2 psia	15.2 psia	------	14.7
Maximum or Relief	42 psig (57 psia)	1500	25	10 psig	15 psig	70 psi	120
Operating Temperature (K)	~5	4-25	4-5	4.2 K	4.2 K	4.2 K	4.2 K
Nominal Mission Length	6 Hr Flight + 8 Hr Standby	5 Days	------	------	------	7 Min Flight + 6 Hr Standby	1 year
Weights/Capacities							
Dry Tankage (lb)	------	115	354	1750	780	11.5	1342
Total Fluid							
(lb)	2.9	48	185	276	153		950
(liters)	10.7	175	670	1000	550	1.04	------
Usable Fluid							
(lb)	2.14	43	------	------	138	------	850
(liters)	7.8	156	------	------	500	------	------
Total Non-Usable (lb)	------	120	------	------	------	------	1442
Total (lb)	------	[illegible]	537	2016	918	------	2292
Weight Ratios							
Dry Tankage/Total Fluid	------	2.4	1.9	6.35	5.1	------	1.41
Total Fluid/Total	------	0.295	0.34	0.134	0.165	------	0.415
Non-Usable/Usable	------	2.8	------	------	5.75	------	1.7
Dimensions	(Cylinder) 9 in. dia. x 10 in. long	(Sphere) 33 in. O.D.	(Cylinder) 36 x 50 x 75 in. long	(Cylinder) 54 in. dia. x 78 in. long	(Cylinder) 44 in. dia. x 72 in. long	(Cylinder) 7.5 in. dia. x 10.0 in. long	(Cylinder) 72 in. dia. x 93.5 in. long
Heat Leakage							
Total (Btu/hr)	~1.0	8	1.1	1.17	0.79	0.34	0.77
Per Unit Area (Btu/hr-ft^2)	0.46	0.136	------	~0.10	------	------	0.005
Loss Per Day (Percent)	------	No Loss Up to 100 hr	1.5 (2.76 lb/day)	1.0 (2.8 lb/day)	1.25 (1.9 lb/day)	------	~0.01
Status	Flight Qualified 1 Suborbital Flight	Production	Operational	Standard Production Item	Standard Production Item	One Flight	Design
Acceleration Loading	------ ------	------ ------	4.5g vertical 8.0g horizontal	10g vertical 5g horizontal	------ ------	------ ------	± 10 g axial ± 3 g lateral

*High Energy Astronomical Observatory



*Supercritical storage systems

Fig. 3-12. Storage Penalties for Cryogenic Oxygen

A system designed under such restrictions utilizing supercritical helium for use over a 60-day period is shown in Fig. 3-13. The vented fluid is used to cool a series of heat loads ranging from 15 to 75 K prior to cooling of the cryogenic tank radiation shields. A constant temperature withdrawal process was used ($T \approx 9$ K) while allowing the tank pressure to decay from an initial value of 240 psia to a final value of 35 psia. Using this approach, many fluids can be utilized at varying temperature requirements to provide long-term cooling at cryogenic temperatures. If high storage pressures are used, the heat content per pound of fluid withdrawn from the tank can be increased substantially. The limitation of this approach is that as the tank is emptied, the temperature of the remaining fluid is increased and may go beyond the desired cooling temperature.

Curves showing the heat content per pound of fluid and the temperature of the fluid as a function of the bulk density of the fluid and the operating pressure can be utilized to establish feasibility and approximate fluid requirements for a given heat load. Thermodynamic data which can be used for this purpose in the general temperature range of 4 to 300 K are presented in Figs. 3-14 through 3-16 for helium, hydrogen and nitrogen based on computations made in Ref. 47.

C. CRYOGENIC SOLID STORAGE

1. BACKGROUND

Development of a cooling system based on the sublimation of a solid coolant into the high vacuum of space shows considerable promise, since a number of problems associated with either subcritical or supercritical storage are avoided. This type of cooling system consists of a solidified gas or liquid, an insulated container, an evaporation path to space, and a conduction path from the coolant to the device being cooled (Fig. 3-17). The operating temperature obtainable with this system depends upon the choice of coolant, the pressure maintained in the system, and the heat load. If the vapor flow rate (which in turn regulates the back pressure and temperature of the effluent flow) is varied, a specific operating temperature can be maintained. The system's operating time depends on the amount of coolant and the heat load.

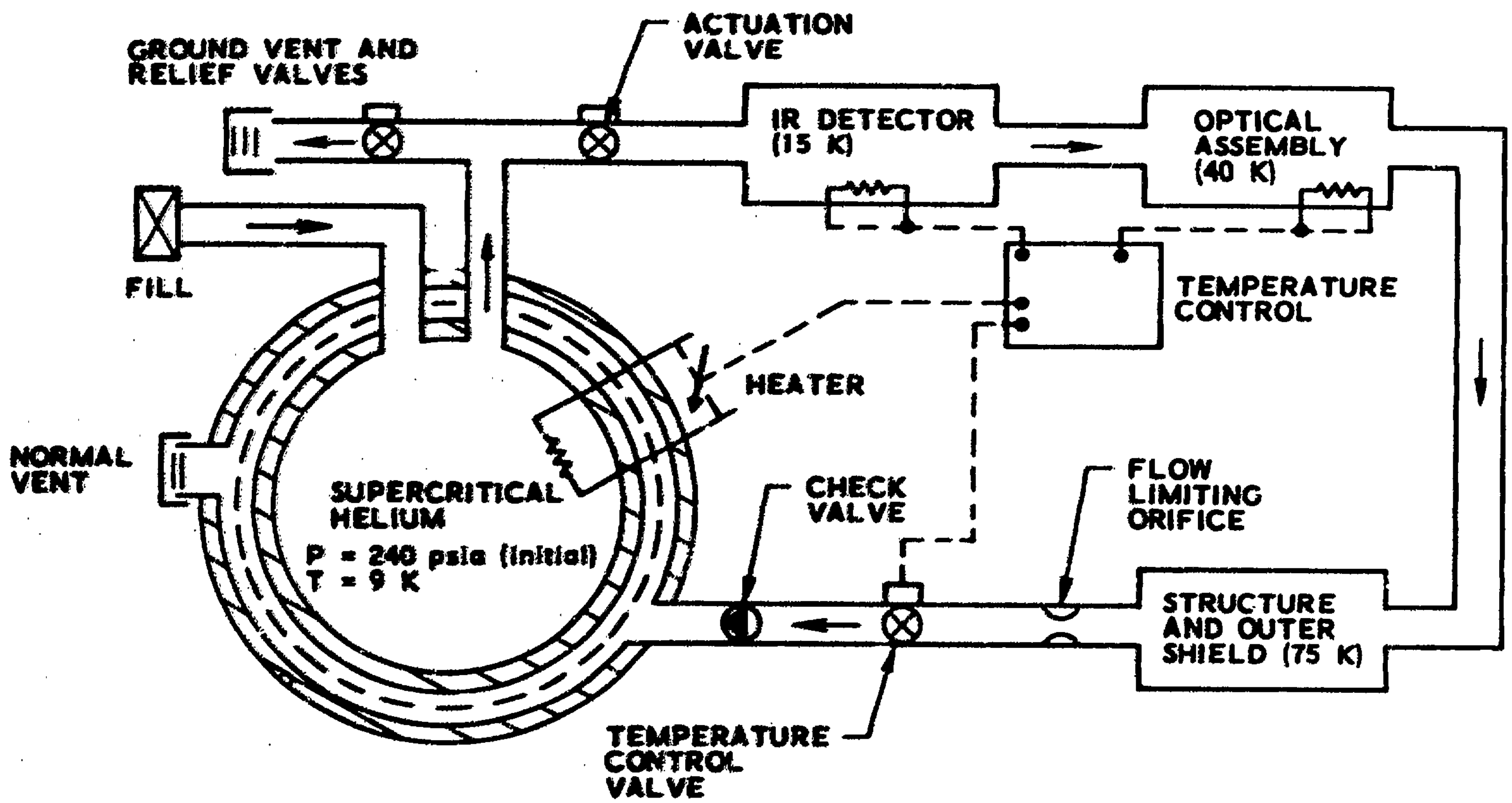
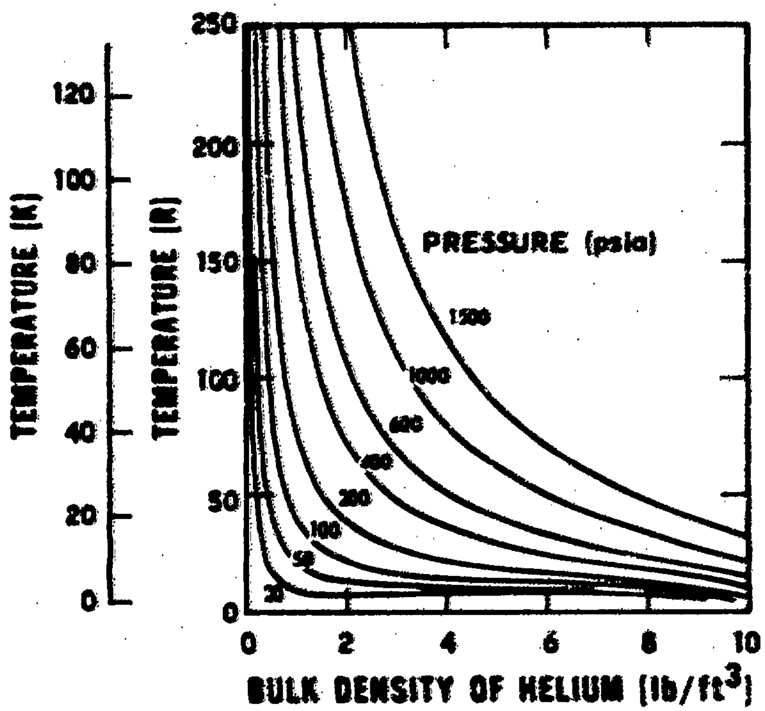
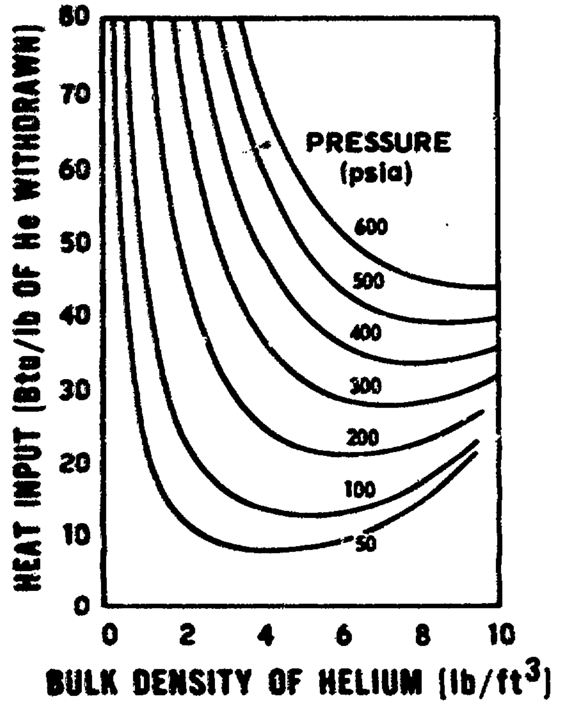


Fig. 3-13. A Hypothetical IR Detector Cooling System Utilizing Supercritical Helium

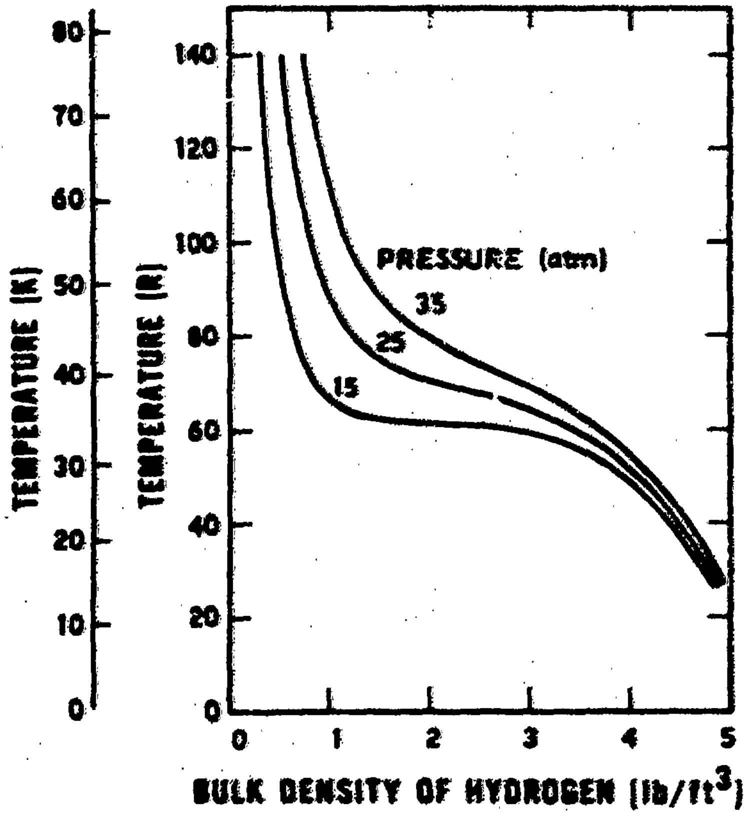


TEMPERATURE vs DENSITY FOR SINGLE-PHASE HELIUM (Ref. 47) (a)

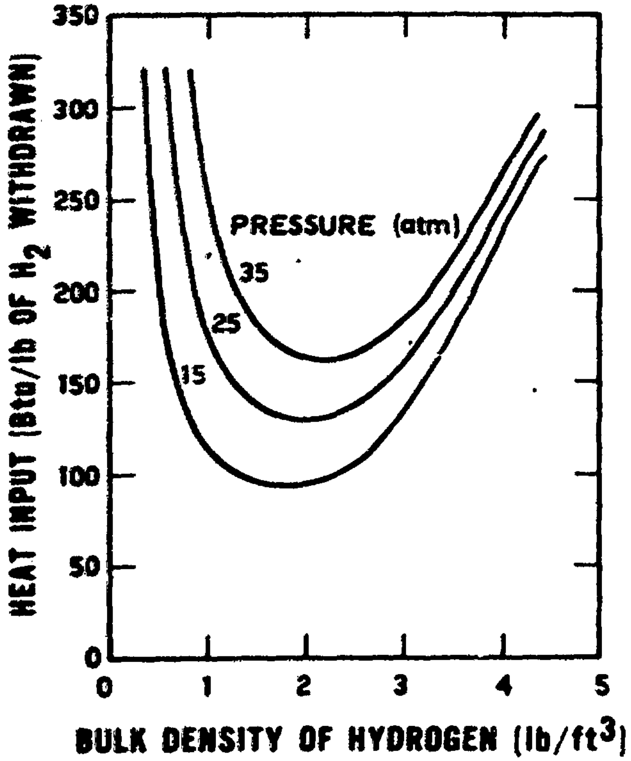


HEAT INPUT vs DENSITY FOR WITHDRAWAL OF SINGLE-PHASE HELIUM (Ref. 47) (b)

Fig. 3-14. Thermodynamic Data for Storage of Single Phase Helium



TEMPERATURE vs DENSITY FOR SINGLE-PHASE HYDROGEN
(Ref. 47)
(a)



HEAT INPUT FOR SINGLE-PHASE WITHDRAWAL OF HYDROGEN
(Ref. 47)
(b)

Fig. 3-15. Thermodynamic Data for Storage of Single Phase Hydrogen

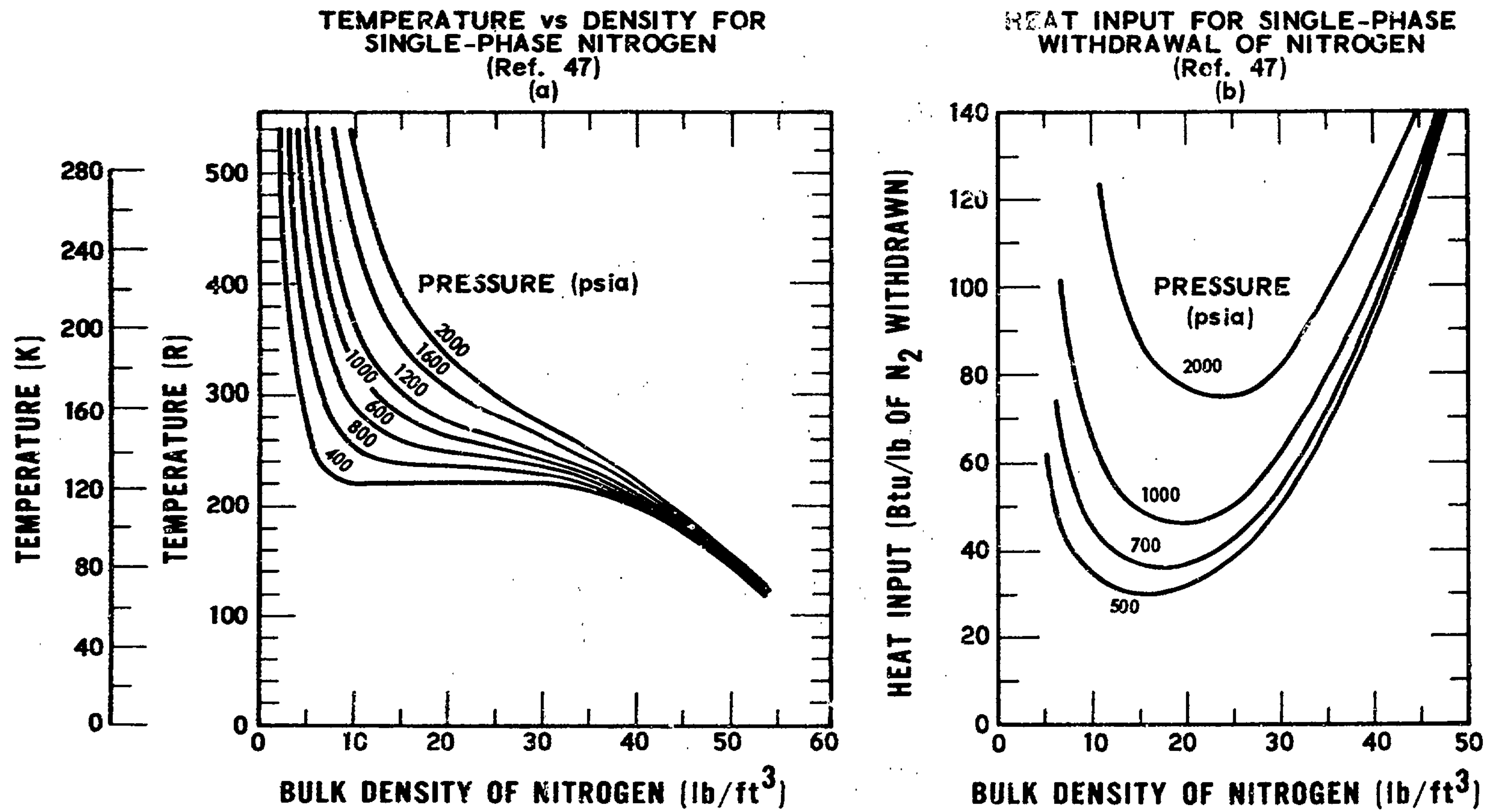


Fig. 3-16. Thermodynamic Data for Storage of Single Phase Nitrogen

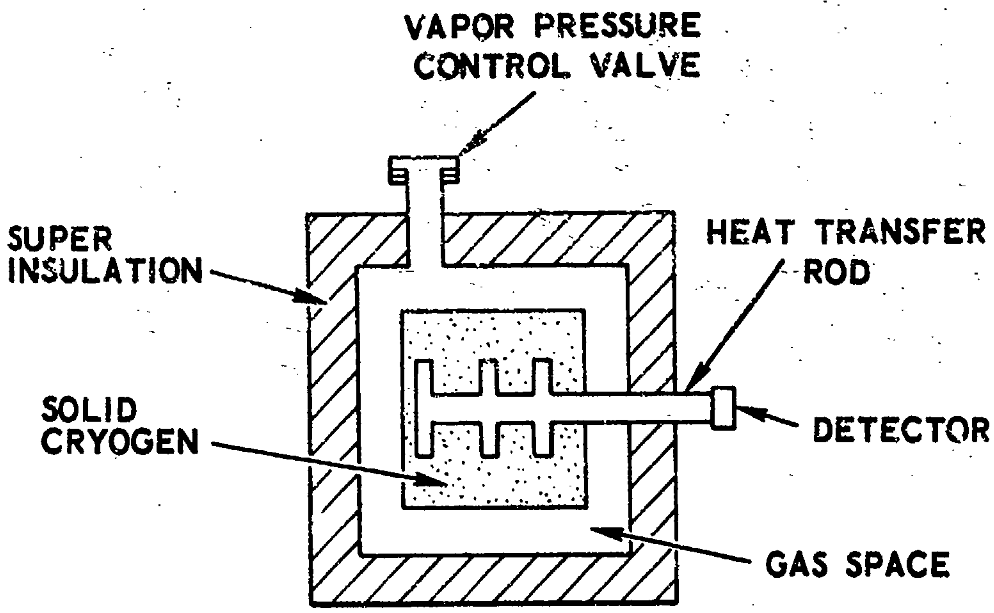


Fig. 3-17. Schematic of Typical Solid Cryogen IR Detector Cooling System

The obvious advantages of the use of a solid system over the liquid include (a) a higher heat content for the heat of sublimation, (b) a higher density storage, and (c) the lower temperature solid phase permits a gain in sensitivity in certain infrared detector systems.

2. SYSTEM OPERATION

The operation of a cryogenic solid system is based on the interrelation of pressure and temperature of a solid in equilibrium with its vapor. Addition of heat sublimes the solid coolant and tends to cause the vapor pressure and thus the temperature to increase. The pressure and the temperature are maintained constant by venting the vapor to space at the appropriate pressure level. The cooling capacity of a given weight of solid coolant is equal to the sum of the heat of fusion and heat of vaporization. Nominal operating temperature and pressure ranges for several solid coolants are given in Table 3-8. Curves of vapor pressure versus temperature are shown in Fig. 3-18. Additional cryogenic data of interest with regard to solids are summarized in Table 3-9. If the temperature requirement of the detector to be cooled is substantially higher than the triple point temperature of the solid cryogen being used, the sensible heat of the effluent vapor can be used to further advantage. In this case, the detector does not have to be mounted directly to the solid via a heat transfer rod but merely exposed to the effluent vapor.

3. SYSTEM DESIGN CHARACTERISTICS

To provide comparative data, the results of analyses of solid cryogenic system weights (from Ref. 33) using hydrogen, neon, nitrogen, carbon monoxide, argon, and methane are given in Table 3-10. These theoretical weights (cryogen and insulation only) are based on an infrared detector heat load of 100 mW for one year of operation at the temperature of the coolant as indicated. The solid cryogen configuration is assumed to be a cylindrical container (length = diameter) with a container external temperature of 300 K.

Table 3-8. Nominal Temperature and Vapor Pressure Ranges of Solid Coolants

Coolant	Temperature Range (K)	Corresponding Vapor Pressure Range (mmHg)
Hydrogen	10 - 14	2 - 56
Neon	16 - 24	1 - 240
Oxygen	48 - 54	0.01 - 2
Nitrogen	47 - 63	1 - 95
Argon	55 - 83	1 - 500
Methane	67 - 90	1 - 80
CO_2	125 - 194	0.10 - 760

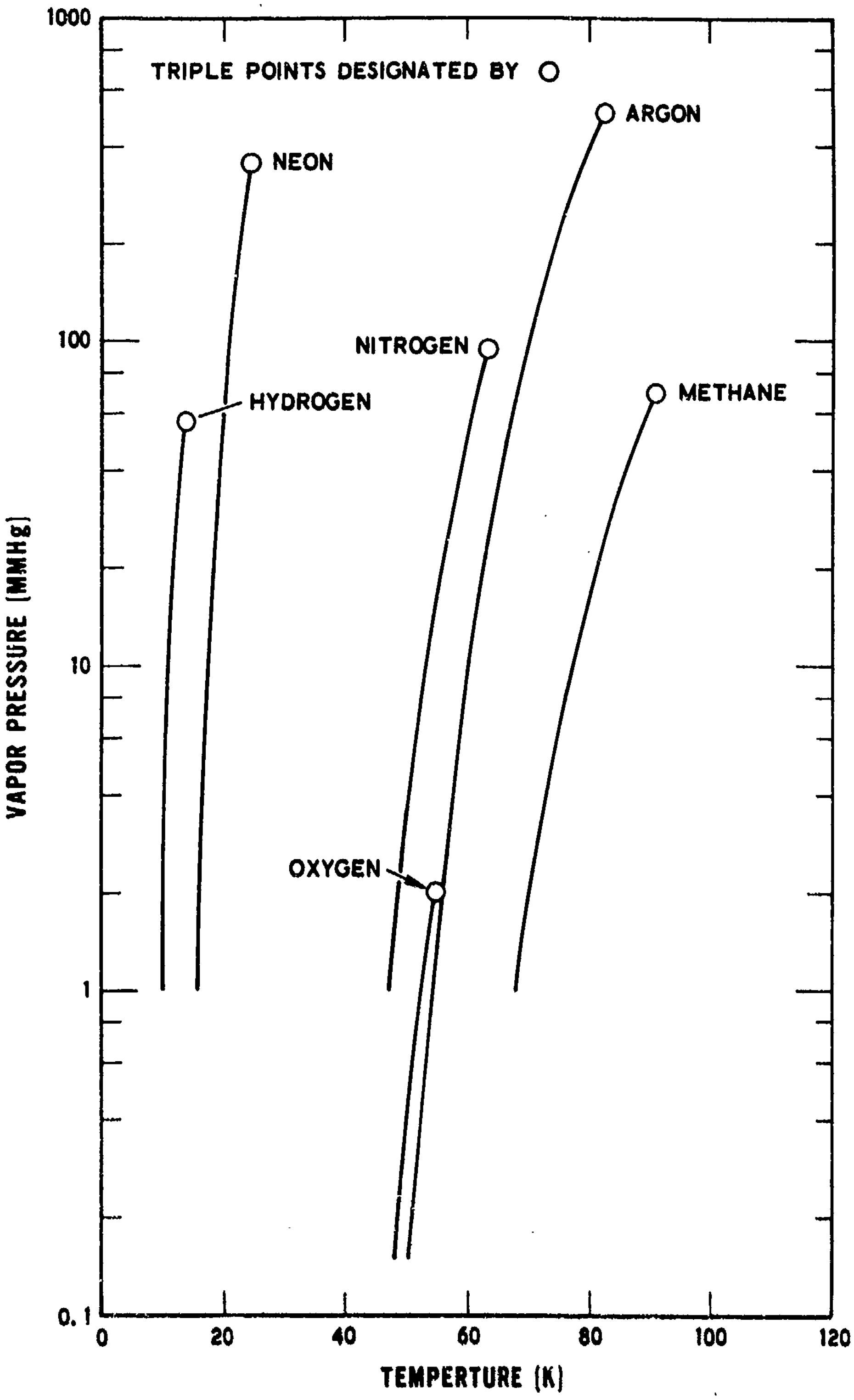


Fig. 3-18. Vapor Pressure of Various Cryogenic Solids

Table 3-9. Cryogenic Properties of Interest for Selected Solid Coolants (Refs. 33, 34, 48, 49, 50)

Coolant	Hydrogen	Neon	Nitrogen	Argon	Oxygen	Methane	Carbon Dioxide
Boiling Point at 1 atm (K)	20.39	27.2	77.4	87.4	90.1	111.7	194.6
Melting Point at 1 atm (K)	13.98	24.5	63.4	83.6	54.4	90.7	215.7
Vapor Pressure of Solid at Melting Point (mmHg)	54.0	323.0	96.5	516.0	2.0	71.0	(75 psia)
Heat of Sublimation (Btu/lb)	218.5	45.5	96.6	79.8	97.5	244.5	246.6
Density of Solid* (lb/ft^3)	5.02	89.8	63.8	107.0	81.3	31.1	97.5
Critical Temperature (K)	33.2	44.5	126.1	150.8	154.1	190.5	304.5

*At melting point

Table 3-10. Estimated Coolant and Insulation Weights for Various Solid Cryogen Coolers (Ref. 33)

- 100 mW Detector Load
- 300 K Exterior Environment
- Cylindrical Container (L = D)

Coolant	Temperature (K)	Weight of Coolant and Insulation (lb)
Hydrogen	12	65.5
Neon	24	119.0
Nitrogen	61	61.5
Carbon Monoxide	68	58.5
Argon	84	66.0
Methane	88	26.2

Total system weight and volume as a function of operating life for hydrogen and methane (which are two of the better solids because of the high heats of sublimation) are shown in Fig. 3-19.

Estimated weights of cryogenic coolers using solid hydrogen as a function of detector temperature and heat load are shown in Fig. 3-20. These weights are based on a simple model utilizing the heat of sublimation as well as the sensible heat of the hydrogen vapor up to the detector temperature.

4. PROTOTYPE OR LABORATORY SYSTEMS

A number of solid cryogen coolers have been developed under NASA and military funding utilizing nitrogen, argon, CO_2, neon, methane, oxygen and hydrogen. The design characteristics of some typical experimental solid cryogen coolers built and tested by Aerojet-General (Azusa, California), Lockheed Missiles and Space Company (Palo Alto, California), and Ball Brothers Research Corporation (Boulder, Colorado) are summarized in Table 3-11. Additional details of some of these units are provided in the following paragraphs.

a. Aerojet-General Solid Nitrogen Cooler

Aerojet performed work under a two-phase Air Force contract during the period 1964-1967 as reported in Ref. 50. Phase I efforts included a parametric analysis of spherical as well as cylindrical cooler configurations, development of a computerized method of optimizing cooler designs, the display of these data in a series of interrelated graphs to facilitate design verification and, finally, the fabrication of an experimental model cooler using solid nitrogen. Phase II entailed performing experiments with the cooler to improve coolant loading and solidification techniques for optimum cooler performance. Included were the studies and experiments relating to thermal conduction through insulation, solids, gases, and various types of mechanical joints.

The experimental solid cryogen (nitrogen) cooler developed in Phase II of this study was designed to provide approximately 0.75 W of cooling

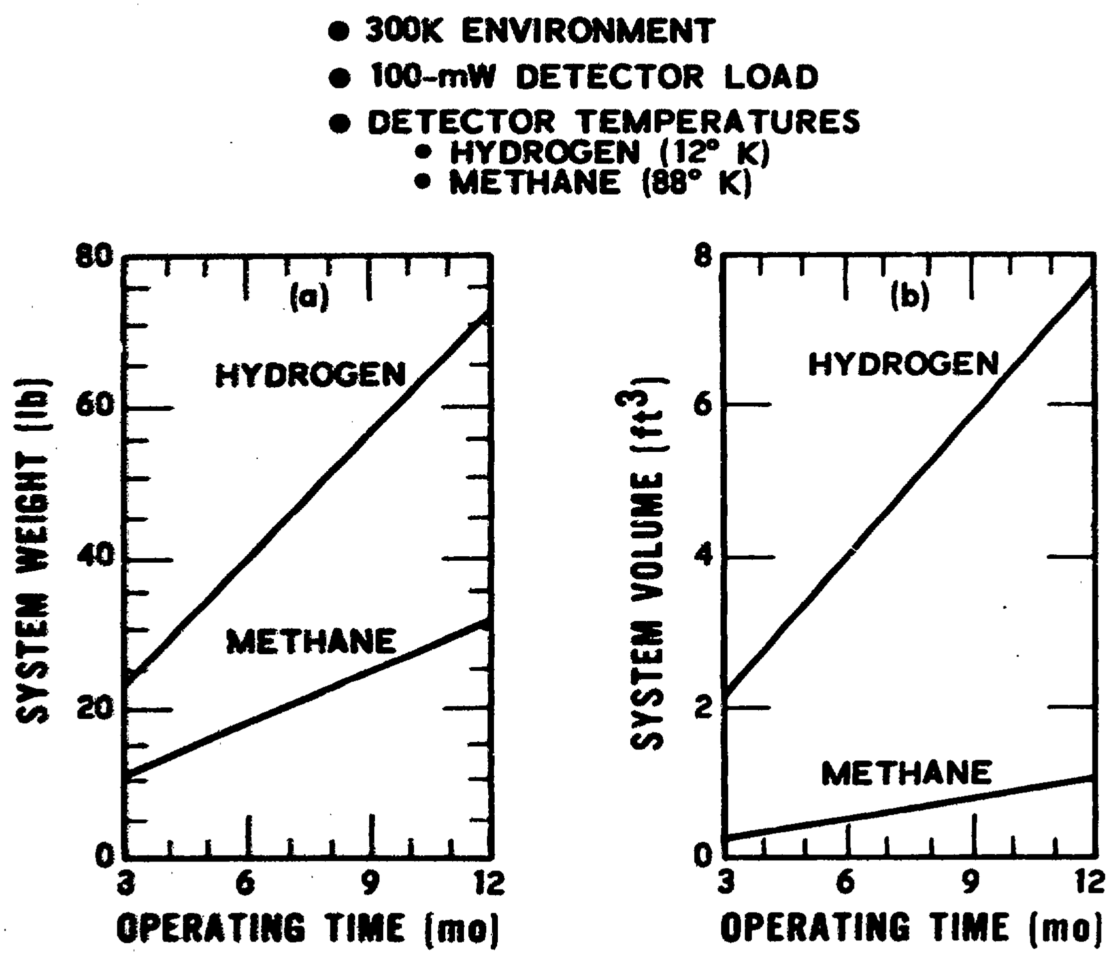


Fig. 3-19. Theoretical System Weight and Volume for Solid Hydrogen and Methane Coolers (Ref. 33)

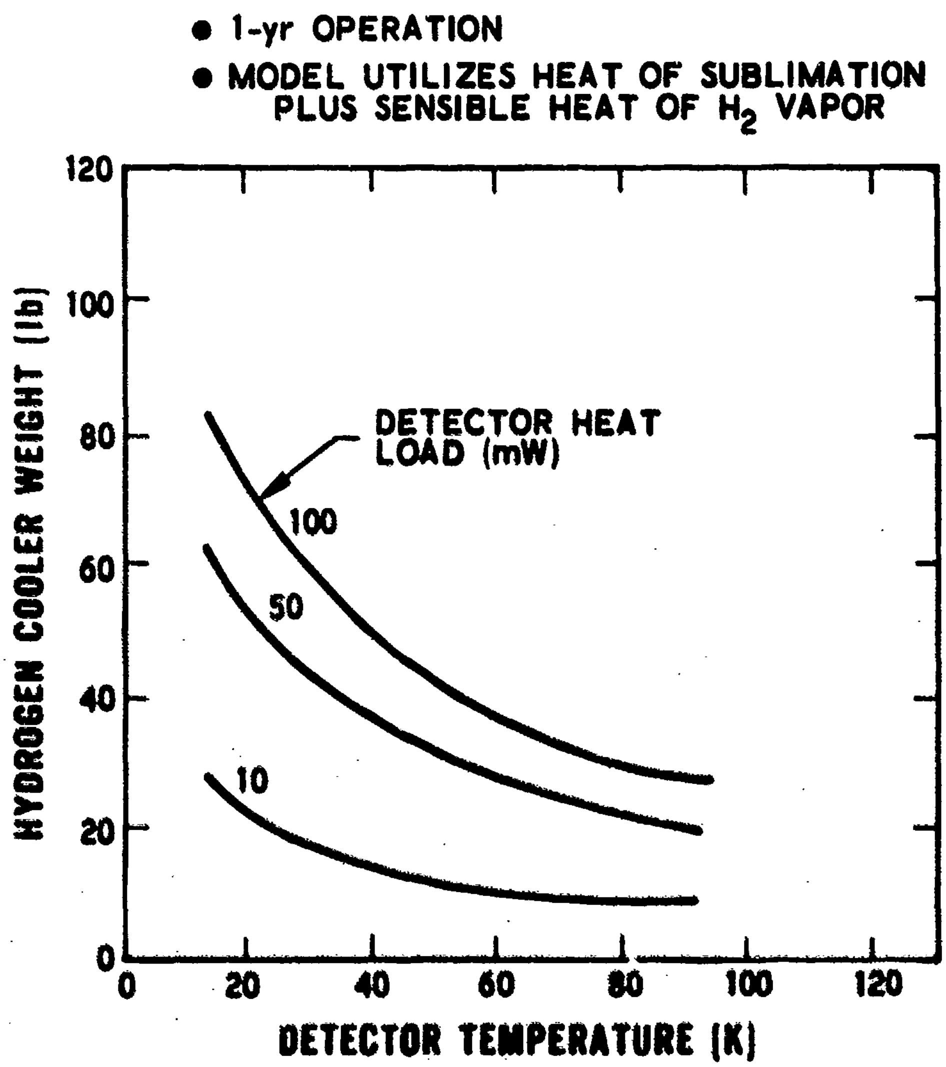


Fig. 3-20. Weight of Solid Hydrogen Cooler Versus Detector Temperature and Heat Load (Ref. 51)

Table 3-11. Summary of Typical Solid Cryogenic Systems

Manufacturer/Developer	Aerojet/General	Lockheed	Aerojet/General	Ball Bros. Research Corp.	Aerojet/General	Aerojet/General	Aerojet/General
Nature of Program	Experimental	Experimental	Design Study	Experimental	Experimental	Experimental	Experimental
Primary Use	IR Detector Cooling	IR Detector Cooling	O_2 Storage	IR Detector Cooling	Airborne IR	IR Telescope	Space Observatory
System Operating Temperature (K)	58	50	12 - 54	76	25	78	10-12
System Operating Pressure (mmHg)	29.4	0.06 - 0.15	0.01 - 2.0	189	------	------	2-3
Design Operating Life (days)	41.6	365	250	365	19	210	78
Primary Heat Load (mW)	750	25	None	14.3††	200	100	200
Total Heat Leak (mW)	310	102.6	200	134.7	------	------	------
Total System Weight (lb)	38	34.1	----	47.4	85	95	150
Refrigerant	Nitrogen	Argon/CO_2	Oxygen	Argon/CO_2	Neon	Methane	Hydrogen
Refrigerant Weight (lb)	27	Argon 13.7 CO_2 8.8	270	Argon 9.02 CO_2 11.62	------	------	------
Configuration	Cylinder	Cylinder	Sphere	Cylinder	Cylinder	Cylinder	Cylinder
System Dimensions	L=D=10.5 in.	L=D=9.25 in.	Dia. = 60 cm	D=13 in., L=22 in.	D=12 in., L=12	D=18 in., L=28 in.	D=30 in., L=33 in.
Effective Density of Refrigerant (lb/ft^3)	63.8	A = 105.5 CO_2 = 103.2	91.2	------	------	------	------
Heat Leakage** ($Btu/hr\text{-}ft^2$)							
Primary Refrigerant	0.75	0.06 (A)	0.53	34.2 mW	------	------	------
Secondary Refrigerant	----	0.148 (CO_2)	----	100.5	------	------	------
Overall	----	0.105	----	------	------	------	------
Standby Time* (days)	4.5	17 (max)	250†	------	------	------	------
Maximum Life with No Load (days)	106	----	525	------	------	------	------
Sponsor of Original Program	AFFDL	NASA	AFAMRL	In-House	Army	NASA	NASA
Current Status of Program	Completed	Completed	Completed	Testing	Completed	Completed	Completed
Current Company-Sponsored Effort	Minimal	None	None	Yes	------	------	------
Reference	50	48	49	52	------	------	------

*Standby time is time required for refrigerant to begin to melt after sealed and vacuum source disconnected.

**No load unless otherwise indicated

†Estimated time for oxygen to melt and boiloff to begin

††Includes IR radiation through optical window

(2.56 Btu/hr) for approximately 1000 hr. The basic configuration is a cylinder of L = D = 10-1/2 in. which contains a theoretical maximum of 32 lb of nitrogen at a solidified density of approximately 63.8 lb/ft^3. The total weight of the system is approximately 38 lb.

"Standby" times for this design have been calculated as 108 hr based on experimental test data. Standby time is defined as the time required for the solid coolant to just begin melting after the vacuum pump is disconnected and the unit is sealed. The unit is normally charged with solid nitrogen at approximately 48 K and 1.5 mmHg and can be allowed to increase to near the triple point of 62 K and approximately 80 mmHg while still maintaining only solid and vapor. Based on the results of the experimental tests, the maximum life was calculated as 106 days for the fully charged (27 lb of solid nitrogen) cooler. Note that the actual solid nitrogen weight is significantly less than the 32-lb theoretical value. This difference is generally attributed to boiloff losses during solidification as well as voids in the solid nitrogen.

The measured heat leak of the entire unit was 0.31 W or 1.06 Btu/hr, not including the detector heat load. To transfer heat from the IR detector to the solid cryogen, a one-half inch diameter solid copper rod extends through the solid. This rod, plus various other connections, fill and vent lines, etc., together with the relatively small overall dimensions, accounts for the relatively high effective heat leak value of 0.75 Btu/hr-ft^2 (based on outside dimensions of the cooler).

b. Lockheed Solid Argon-CO_2 Cooler

Under contract from NASA/Goddard Spacecraft Center, Lockheed Missiles and Space Company, Palo Alto, California, developed a prototype spacecraft solid cryogen refrigerator capable of providing 25 mW of cooling for an IR detector at approximately 50 K (Ref. 26). The objective of the program was to provide cooling for a period of one year based on an external refrigerator temperature of 300 K.

Solid argon (maintained at 50 K and approximately 0.15 mmHg vapor pressure) provides the primary cooling while solid carbon dioxide (CO_2) in conjunction with surrounding evacuated multilayer insulation is used to minimize the heat leak to the argon. The CO_2 is maintained at approximately 125 K. The entire system weighs approximately 30 lb including 22.5 lb of solid cryogen (13.7 lb of argon and 8.8 lb of CO_2). A schematic of the cooler is shown in Fig. 3-21. Based on measured volumes of containers, the bulk densities attained in the cooler were 99.7 and 97.3 percent of the theoretical densities of argon and CO_2, respectively.

The system was designed for a total heat leak to the CO_2 of 76 mW (0.26 Btu/lb) and 40 mW (0.135 Btu/hr) to the argon including the 25 mW IR detector load. The initial tests of the unit resulted in heat loads approximately three times the predicted values. After considerable redesign, improved insulation and reconstruction, the final thermal tests were considerably improved but still short of the design goals. The final measured heat leak was 74 mW (0.25 Btu/hr) to the CO_2 and 28.6 mW (0.098 Btu/hr) to the argon not including the IR detector load. In order to permit the one-year operation, the IR detector load must be reduced to 17.6 mW.

Based on the inside dimensions of the container, the overall effective heat leak per unit area was calculated as follows:

Item	Q/A (Btu/hr-ft^2)
CO_2	0.148
Argon (without detector load)	0.06
Argon (with detector load)	0.112
Total (without detector)	0.105
Total (with detector)	0.132

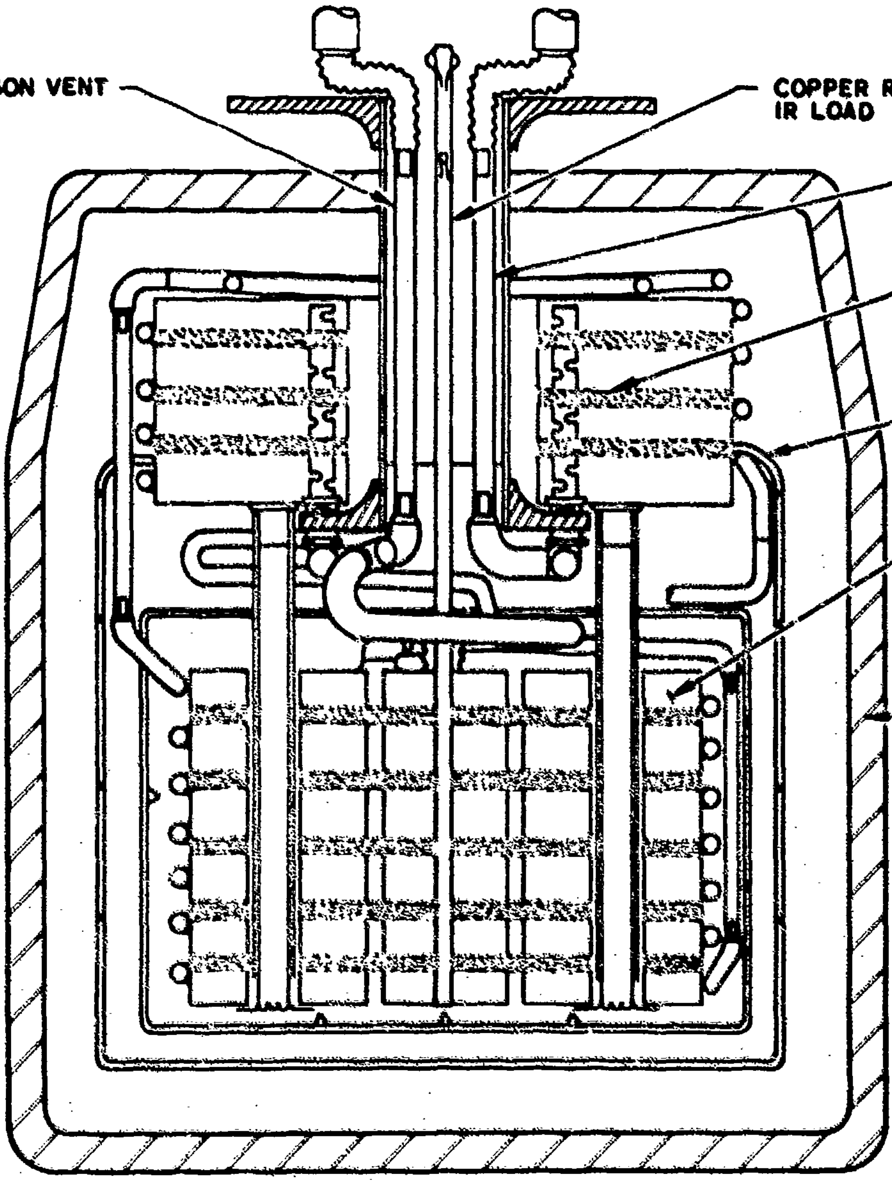


Fig. 3-21. Schematic of Solid Argon-Solid Carbon Dioxide Infrared Detector Refrigerator (Ref. 48)

c. Cryogenic Solid Oxygen

Cryogenic solid oxygen has been considered as an attractive potential method of long-term oxygen storage in space. A design study was conducted by Aerojet-General for the USAF Aerospace Medical Research Laboratories (AMRL) as reported in Ref. 49. The highlights of their studies are summarized below.

Solid oxygen exists in three phases: Phase I between 43.8 and 54.4 K (the triple point), Phase II between 24 and 43.8 K, and Phase III below 24 K. In addition to an increase in the solid density at the lower temperatures, there is an increase in the heat of sublimation. Also, there is a significant heat of transition between Phases II and I and between Phases III and II--resulting in added heat capacity if the solid oxygen is sub-cooled to these lower temperatures. The heat of sublimation for oxygen in these regions is shown in Fig. 3-22.

Studies made in Ref. 49 covered the total storage times for various forms of oxygen including solids, liquid and supercritical states. The results of the studies for a 30-cm radius sphere with 1.0 cm of superinsulation and designed for a 10-g launch load are shown in Fig. 3-23 which indicates the relative amount of oxygen remaining in the tank versus time. The insulation consists of 80 layers of aluminized mylar with an overall density of 0.09 g/cm^3 evacuated to a pressure of 10^{-7} mmHg.

The curves show the significant improvement in storage life in going from a liquid to a triple point solid and then further to the lower temperature solid oxygen phases. The additional advantage of going to below the Phase II solid (i.e., below 43.8 K) represented by curve 3 of Fig. 3-23 does not appear justified because of the additional ground-cooling problems.

Oxygen stored initially as a solid at the triple point can be stored for approximately 130 days longer than oxygen stored as a subcritical liquid, as shown by curve 2 versus curve 1 of Fig. 3-23. The corresponding figures for Phase II solid at 43.8 K and Phase III solid at 12 K (curve 4) are 210 and

250 days when compared to the saturated liquid. Figure 3-23 also shows that after 250 days of storage, a relative weight of 100 percent (i.e., relative to liquid oxygen) remains in the vessel stored initially at 12 K (curve 4).

d. Ball Brothers Research Corporation Solid Argon-CO_2 Cooler

A solid cryogenic cooler utilizing argon and CO_2 for cooling of an infrared detector for up to a year was designed and fabricated by Ball Brothers Research Corporation (BBRC) as an in-house effort. The system was designed to provide for cooling of a detector load of approximately 14 mW at 77 K for up to one year and to meet the Apollo vibration criteria.

Argon, the primary refrigerant, is maintained at 76 K while CO_2, the secondary refrigerant, is maintained at approximately 130 K and acts as a heat barrier for the argon. The envelope of the unit is a cylinder approximately 13 in. in diameter and 22 in. long. The total loaded weight of the system is 47.4 lb with a stainless steel outer shell. BBRC estimates that the weight will be around 37 lb if the stainless steel shell is replaced with aluminum. Both refrigerants are condensed and frozen in their containers by passing liquid nitrogen through a cooling loop which is an integral part of the structure. The multilayer insulation is evacuated to 10^{-6} mmHg, and an ion pump is required to maintain this vacuum for extended periods on the ground.

The measured heat leaks during preliminary testing were in the order of 10 to 20 percent higher than the design heat leak of 134.7 mW. Based on the existing design, a lifetime of about 200 to 250 days appears maximum. Further testing and modification of this system is continuing by BBRC.

D. AMBIENT TEMPERATURE LIQUID STORAGE SYSTEMS

The ability to provide a long duration storage system with a short use time capability is frequently a desirable goal for cryogenic cooling of IR detector systems. Cryogenic liquids or solids require sophisticated insulation systems and provide relatively limited storage time capability. A limited number of liquids exist, however, which can be stored at ambient

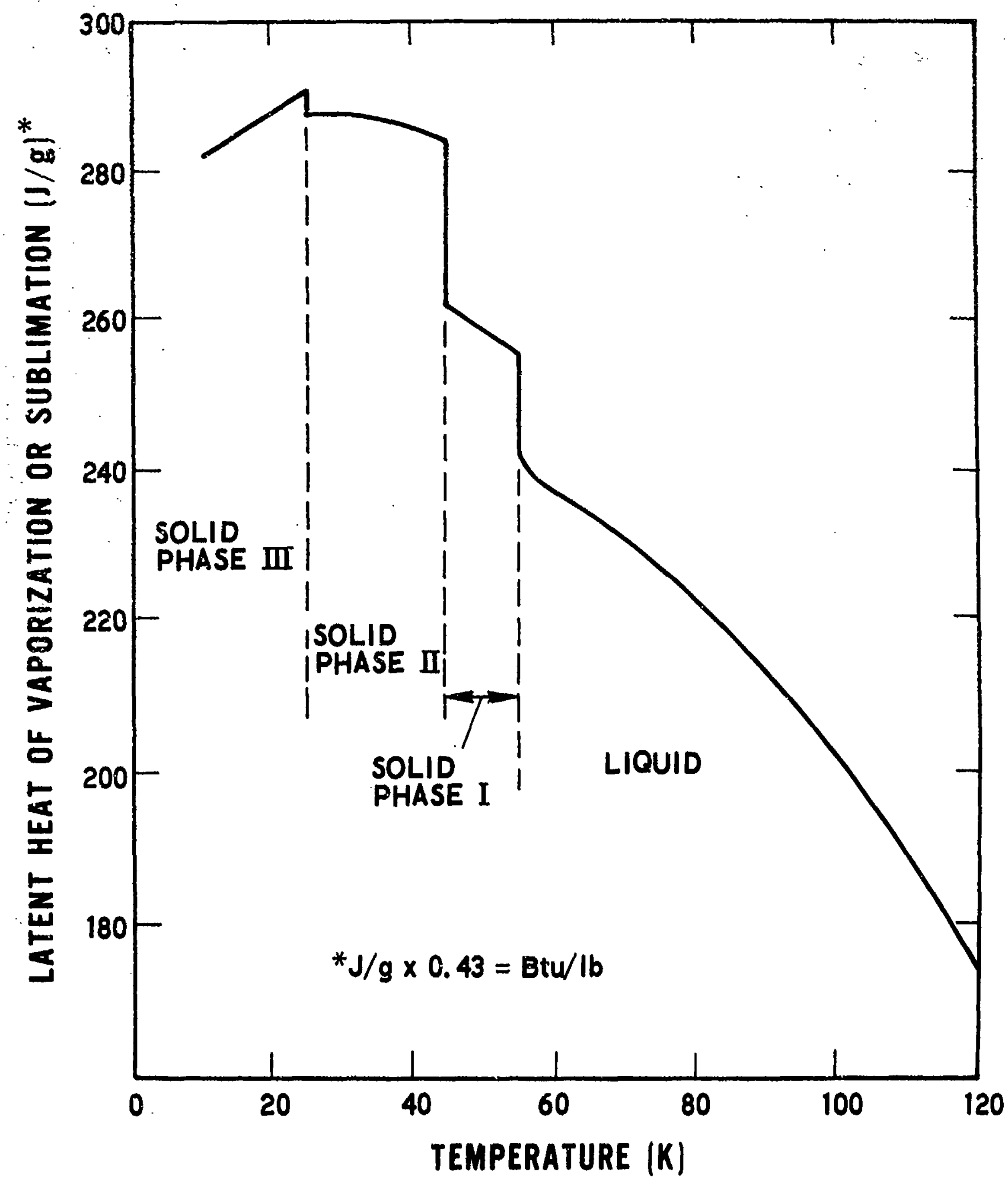


Fig. 3-22. Latent Heat of Vaporization or Sublimation of Oxygen as a Function of Temperature (Ref. 49)

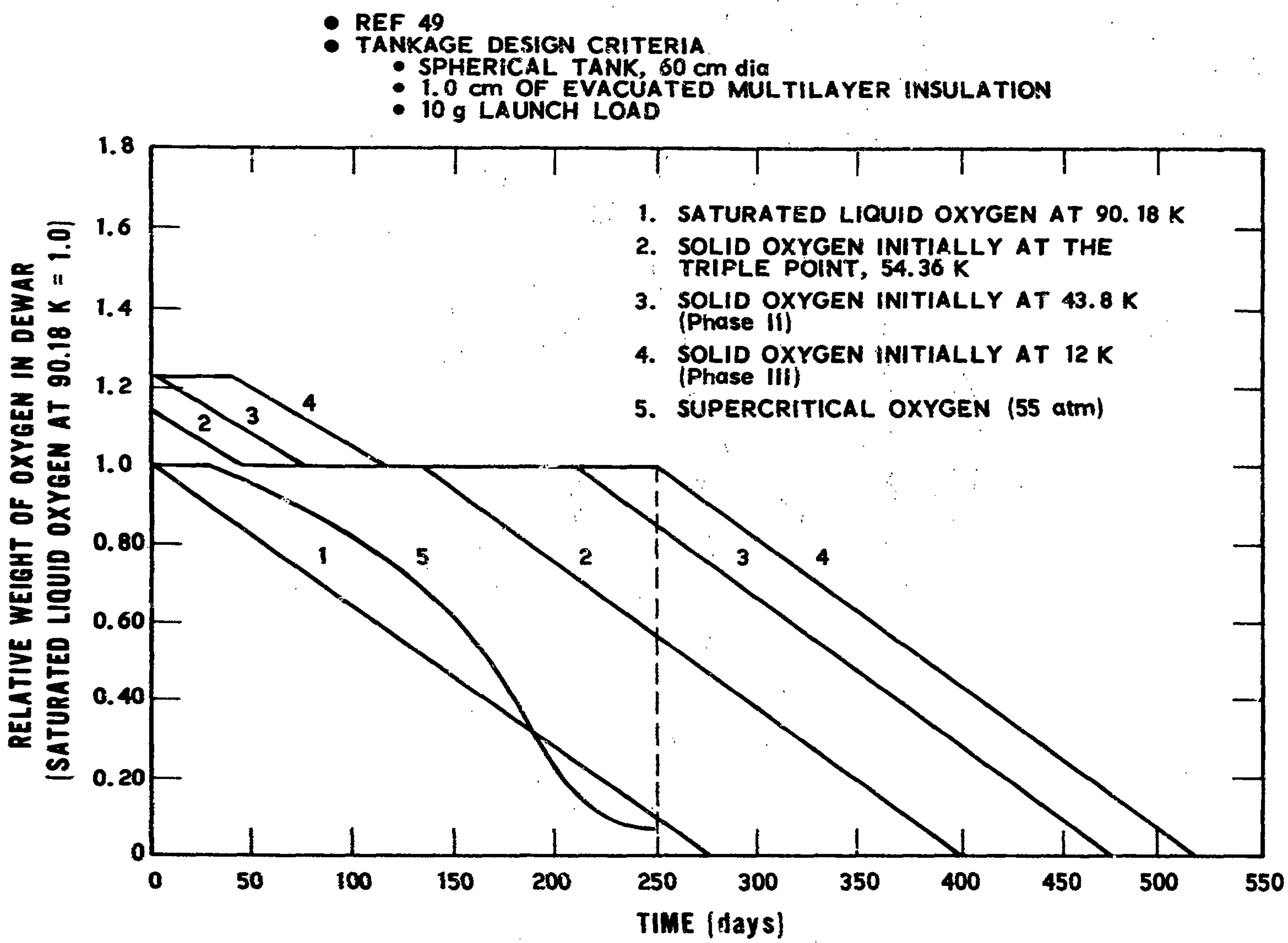


Fig. 3-23. Comparison of Solid Versus Liquid Oxygen Storage

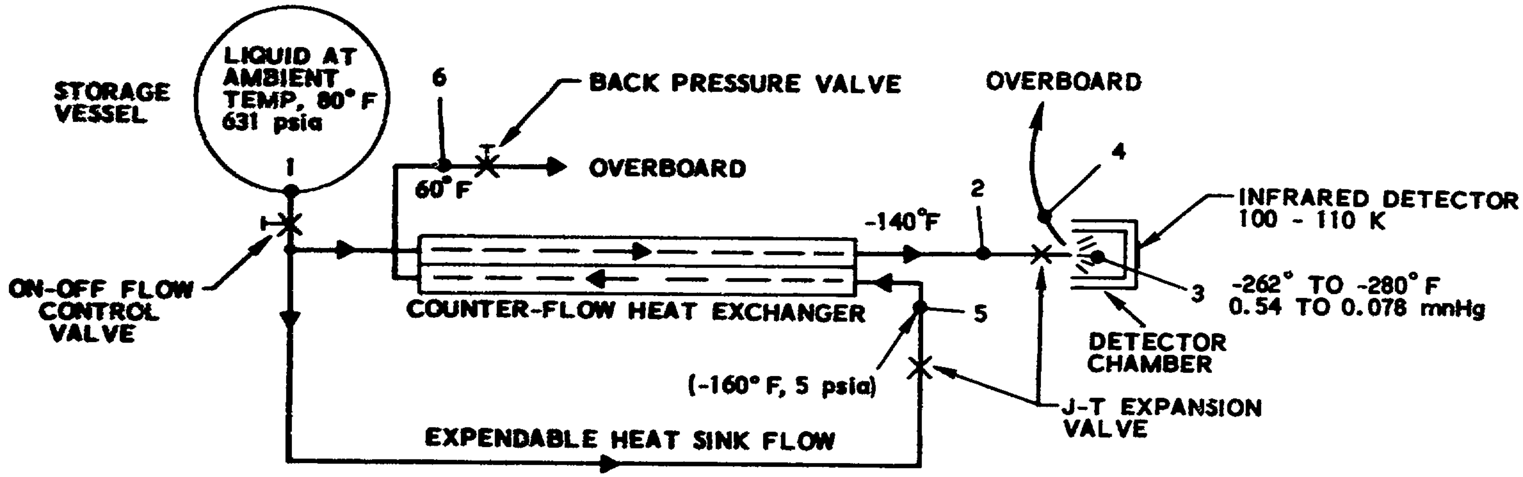
temperature and manipulated thermodynamically to potentially provide a feasible cryogenic cooling system.

Some preliminary analyses have been completed regarding cryogenic cooling with these types of fluids (Ref. 53). The thermodynamic analysis performed in the above study indicated it is feasible to use ethane or possibly propane for cooling of infrared detectors at a temperature level of 100 to 110 K. These two fluids can be stored indefinitely as liquids at ambient temperature (at 80 F the storage pressures are 631 psia for ethane and 144 psia for propane) and manipulated thermodynamically to obtain the desired 100 to 110 K temperature level for cooling.

The cooling system consists of a storage vessel, a small counterflow heat exchanger, two J-T expansion orifices, one flow control valve and a back pressure valve as shown schematically in Fig. 3-24 using ethane as the working fluid. The corresponding thermodynamic process as depicted by a pressure-enthalpy diagram is shown in Fig. 3-25. Assuming 100 percent fluid utilization (all the fluid sprayed in the chamber is vaporized by heat from the detector), the total wet weight of an ethane cooling system is about 10 lb for a total heat load of 1/2 W and a total accumulative operating period (either intermittent with long standby periods or steady) of seven days. A 50 percent utilization efficiency at the detector would result in a system wet weight of about 20 lb.

1. PHASE SEPARATION AND EXPULSION OF LIQUID

Several methods have been considered for expelling and separating the liquid from the storage container. These methods also can be applied to cryogenic liquids to a limited extent. One method is to use a capillary separation device, such as a spherical wrap of screen, around the inside of the storage vessel (see Fig. 3-26a). Liquid would then be withdrawn from the annular area between the screen and the vessel wall. Surface tension forces in the liquid would prevent passage of any gas bubbles from the interior of the vessel across the screen surface. Pressure for expulsion would be supplied by evaporation of the liquid in the interior of the vessel. This concept can be used on a partial basis where only enough fluid is retained at the tank outlet to supply, for example, an engine during restart.



SYSTEM REQUIREMENTS

STORAGE TIME:	30 DAYS TO SEVERAL YEARS
DETECTOR TEMP:	100 - 110 K
COOLING LOAD:	½ W
OPERATING TIME:	7 DAYS

Fig. 3-24. Ambient Temperature Liquid Storage System (Ethane) (Ref. 53)

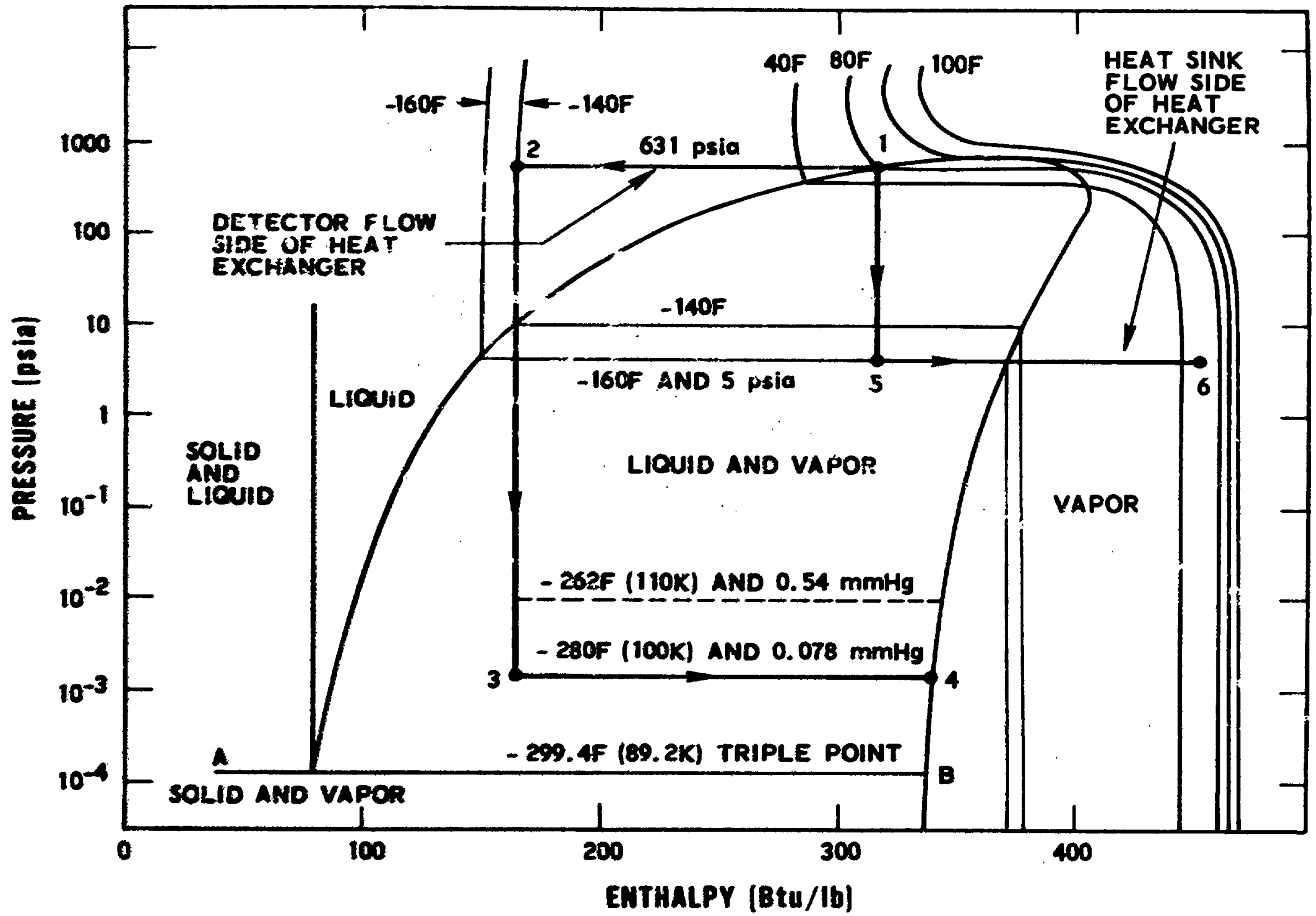
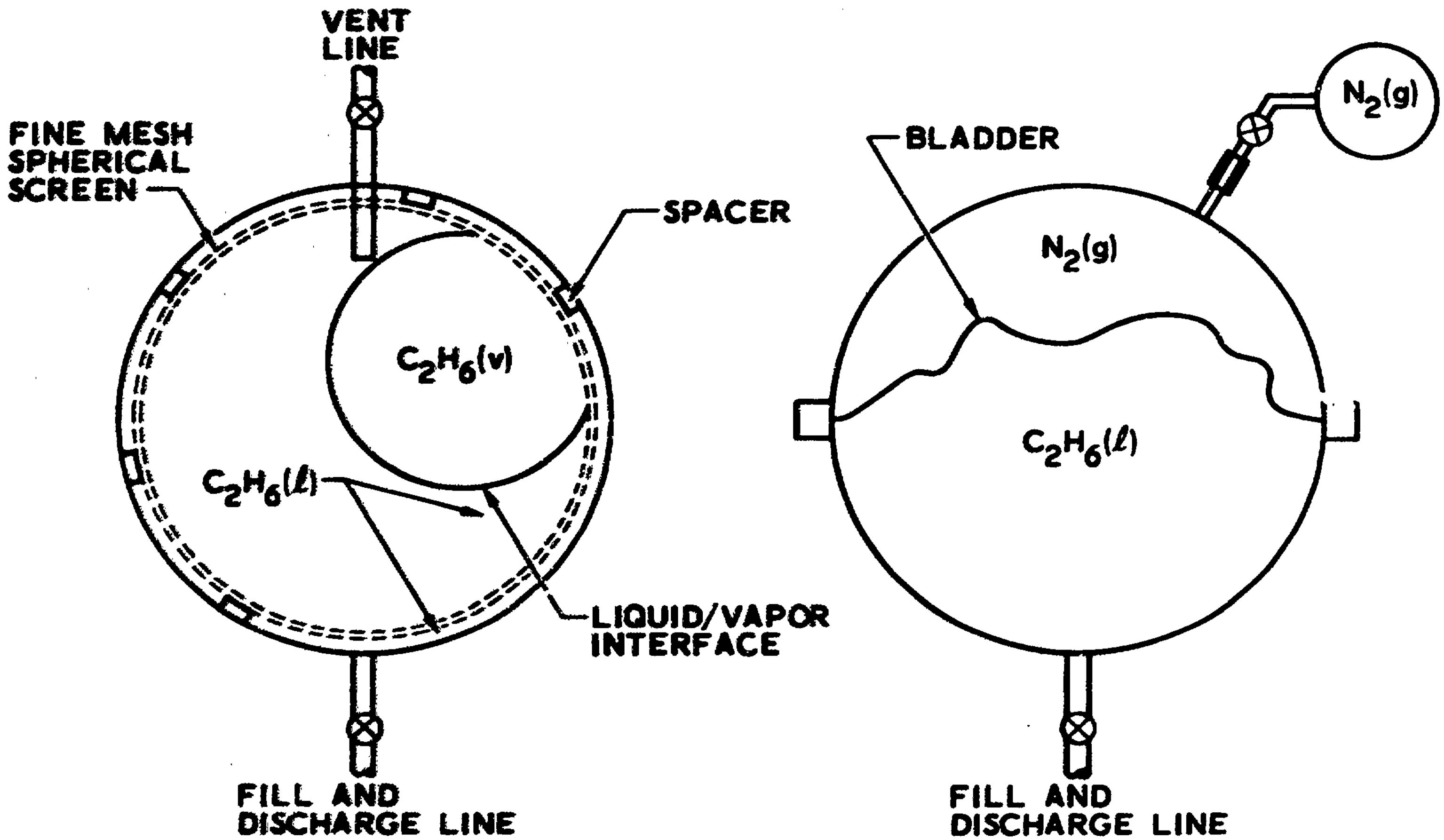


Fig. 3-25. Thermodynamic Process Path for Proposed System Using Ethane as the Working Fluid (Ref. 53)



(a) THERMAL PRESSURIZATION (capillary screen)

(b) NITROGEN GAS PRESSURIZATION (bladder)

Fig. 3-26. Methods of Coolant Expulsion - Ambient Temperature Liquids (Ref. 53)

Capillary system designs have been flown on a number of spacecraft and upper stage propulsion systems including the Agena, the Apollo Service Module (SM), and the Titan III Transtage, as well as on various drone aircraft such as the Northrop MQM-74A and Ryan Firebee II. The Apollo SM (Ref. 54), Agena (Ref. 55), and Titan III Transtage (Ref. 56) use variations of the basic partial retention system wherein capillary screen barriers are utilized to retain liquid over the tank outlets at all times. The previously mentioned target drones also use similar concepts (Refs. 57 and 58). The designs utilized in these programs are illustrated in Fig. 3-27. A total retention system concept was designed and fabricated for the X-15 flight research vehicle hydrogen peroxide tank but was never flight tested.

Another, more common, means of expelling the liquid is to use a bladder (see Fig. 3-26). In this case, high-pressure nitrogen gas regulated to the desired pressure is used to expel the liquid which is kept above the saturation pressure so that no vapor is present in the storage vessel. Although the weight estimates for this system are not shown here, the net result is that system total weight is approximately equal to that of the capillary expulsion system. The reason for this is that a smaller amount of fluid is required (all of the fluid would be utilized rather than ending up with a vessel full of a high density vapor). This compensates for the added nitrogen gas pressure vessel weight.

Bladders have been used on a number of programs. One of the earliest uses was on the X-15 liquid nitrogen tank and the hydrogen peroxide tank. Repeated failures of the mylar bladder in the liquid nitrogen tank, however, resulted in replacement of the bladder with a compartmented tank. A large number of bladder systems have been very successful in conjunction with storable propellants such as in reaction control systems (RCS) for the Apollo Command, Service and Lunar Modules (Ref. 60) and the Surveyor vernier propulsion system (Ref. 61). The bladder materials used in both of these programs are basically teflon compounds consisting of laminated layers of TFE (polytetrafluorethylene) and FEP (flourinated ethylenepropylene) with a total thickness of 0.006 in. The primary limitation of these systems is the bladder life.

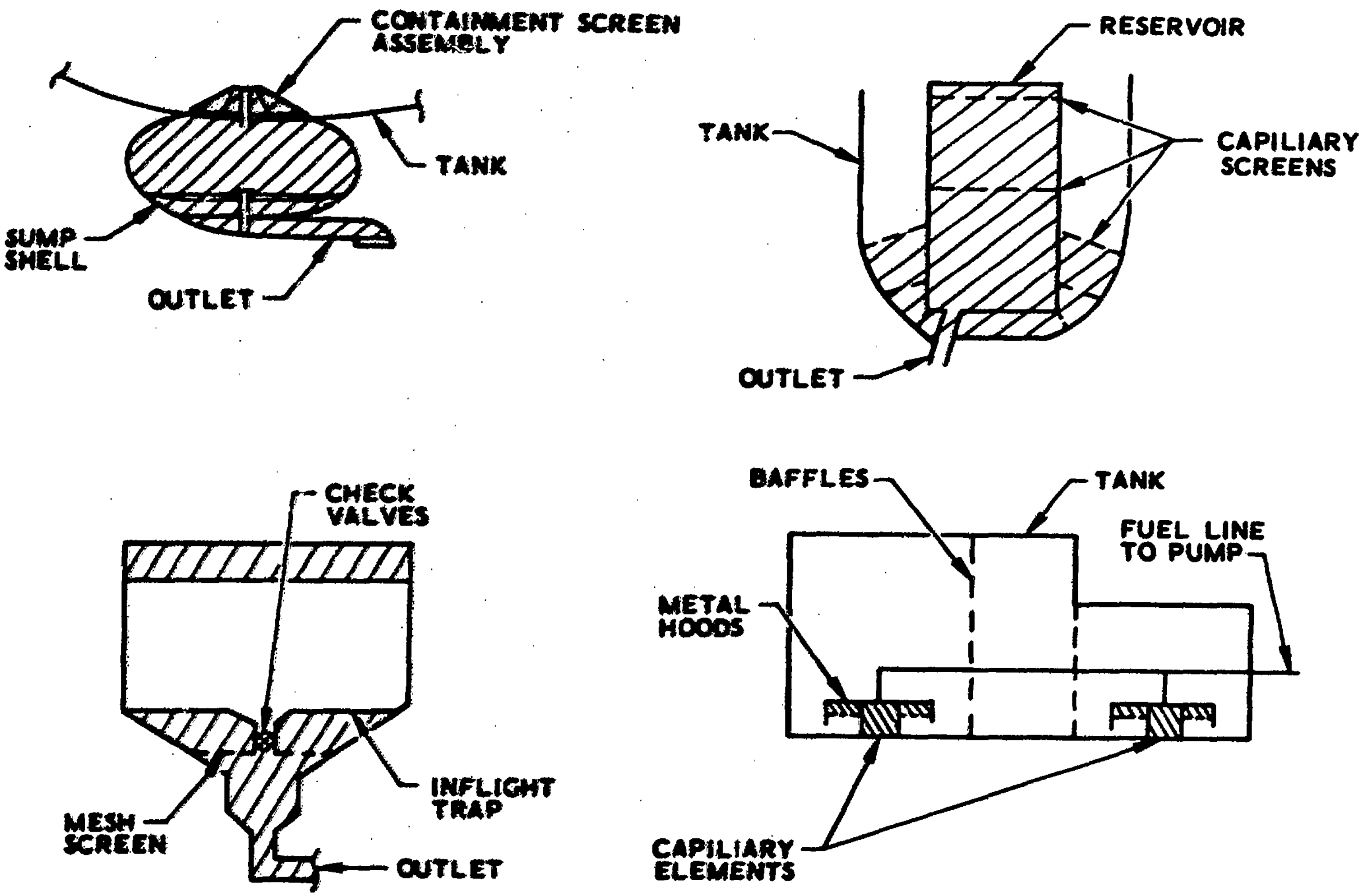


Fig. 3-27. Examples of Capillary Partial Retention System Designs (Ref. 59)

In addition to capillary screen and bladder expulsion devices, a number of other techniques for liquid expulsion have been proposed such as the use of pistons, diaphrams, metallic bellows, and the use of polarization forces (dielectrophroetic devices). Discussion of these techniques is beyond the scope of this report, and the literature (Ref. 62, for example) should be consulted for further details.

The above background data and a recent survey of capillary retention devices (Ref. 59) indicate that adequate design concepts, performance analysis and flight experience exist to establish the feasibility of using either capillary retention systems or bladders in short duration missions. However, because of the lack of established long-life materials compatibility accelerated testing techniques, accurate predictions of long duration mission capabilities of a year and beyond cannot be made. In addition, the materials compatibility with any new fluid would have to be reexamined.

IV. RADIATORS

A. BACKGROUND

Another method of developing cryogenic temperatures in space is to utilize the low temperature sink of space directly by using a cryogenic radiator. This concept is potentially attractive since such a system is completely passive, requires no power, and may be capable of high reliability for extended periods. Recently, there has been considerable activity in the design of such radiators to maintain the temperature of detectors in electro-optical systems at temperatures in the 70 to 150 K region.

The effective temperature of deep space is approximately 2 to 4 K, and a suitably sized cold plate of high emittance to which one or more detectors are mounted can be made to radiate to this sink. It is necessary to shield the cold plate against heat inputs from direct sunlight and, in the case of near-earth orbits, the heat inputs from direct thermal emission and reflected sunlight from the earth and its atmosphere.

Furthermore, the cold plate must be thermally shielded from the parent spacecraft. These considerations usually result in a passive cooler design which is tailored to a particular spacecraft system. The type of orbit (e.g., near-polar, equatorial), orbit altitude, orientation of the spacecraft relative to the earth or sun, and the location of the radiator all significantly influence the design of the radiator.

B. THEORETICAL PERFORMANCE OF PASSIVE RADIATORS

The minimum theoretical area required as a function of the total power dissipated by the radiator and the radiator temperature level is shown in Fig. 4-1 for a radiator of unity emittance, perfectly isolated from any background heat inputs, and radiating to a 0 K sink. The radiating area of a practical system will be larger than that indicated in the figure because of parasitic heat flow, and the area of the complete radiator assembly will be

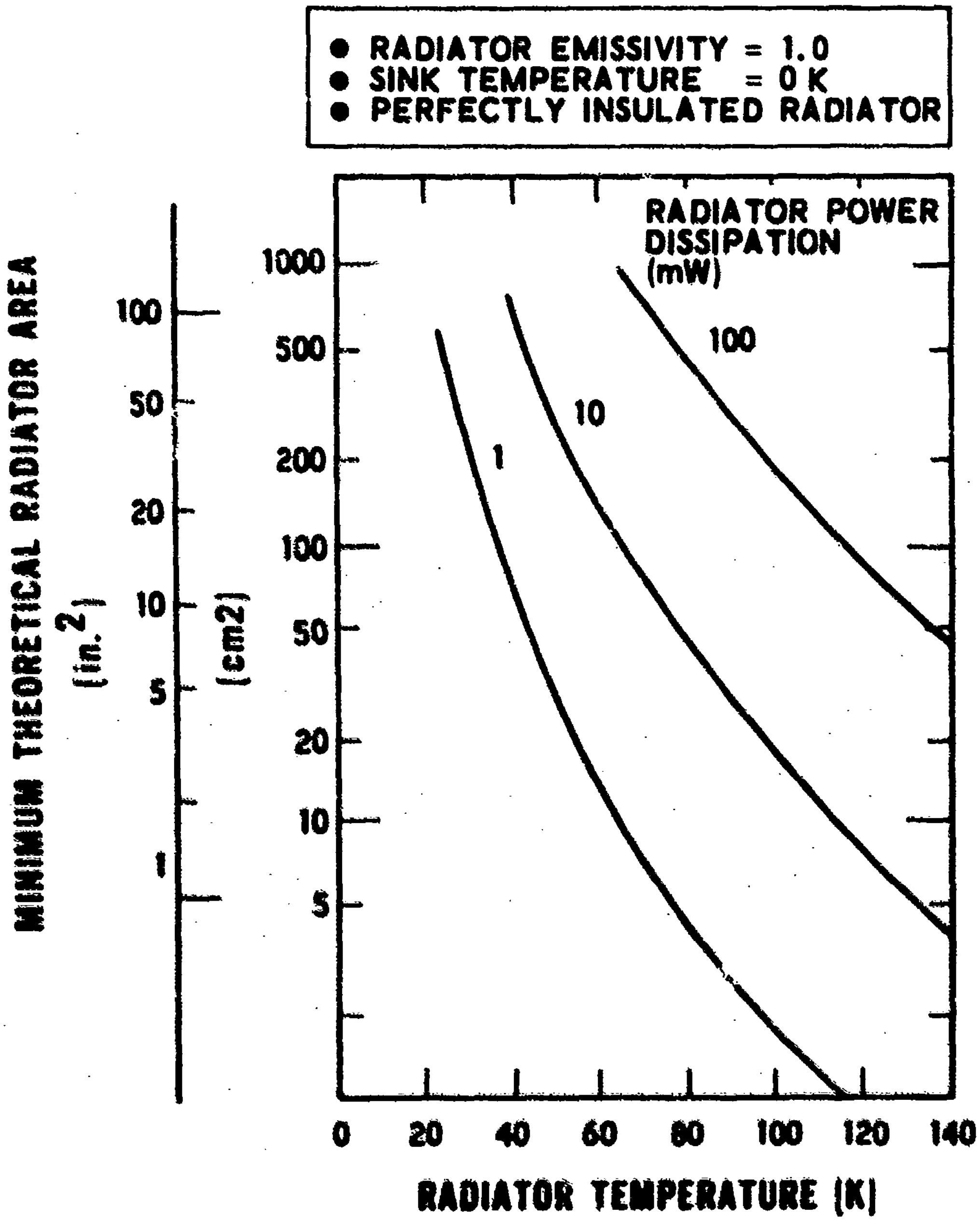


Fig. 4-1. Minimum Theoretical Radiator Area as a Function of Temperature and Heat Load (Ref. 51)

still larger because of the size of the thermal shielding system and structure. More realistic radiator areas required as a function of the radiator temperature and the radiator parasitic heat leak are estimated in Fig. 4-2. These data are based on parametric analyses (Ref. 63) of a hypothetical system. The parasitic heat leak represents the heat transfer between the radiator and the supporting structure and vehicle and is estimated to range from 0.15 to 0.30 W/ft^2 of radiator surface. The curve labeled as zero parasitic heat leak represents a perfectly insulated radiator. The detector system heat load as used in this analysis represents all other heat loads associated with the actual IR detector including the radiation load, power losses, heat leaks due to lead wires, and other miscellaneous heat leaks into the detector assembly.

C. RADIATOR DEVELOPMENT PROGRAMS

Studies on the development of cryogenic radiators for cooling IR systems have been funded by the AFFDL (to Philco-Ford Corporation) and by NASA Goddard Space Flight Center [to International Telephone and Telegraph (ITT) and to Santa Barbara Research Center]. The description and results of these studies are discussed in the following paragraphs.

1. AIR FORCE DEVELOPMENT PROGRAMS

A recent study on a passive radiative cooled system for IR detectors was completed by Space and Reentry Systems Division of Philco-Ford Corporation for the AFFDL, Wright-Patterson AFB, Ohio (Ref. 64). The objective of this program was to study the feasibility of and to develop design techniques for coolers capable of reaching operating temperatures of 77 K (138 R) with nominal heat loads of 10 mW for three different earth orbits as follows:

a. An earth-oriented 200 nmi circular orbit in the equatorial plane

b. An earth-oriented 600 nmi orbit with an orbit plane-sun angle of 45 deg

c. A synchronous earth-oriented equatorial orbit

The maximum radiator size was restricted to a cube 18 inches on a side.

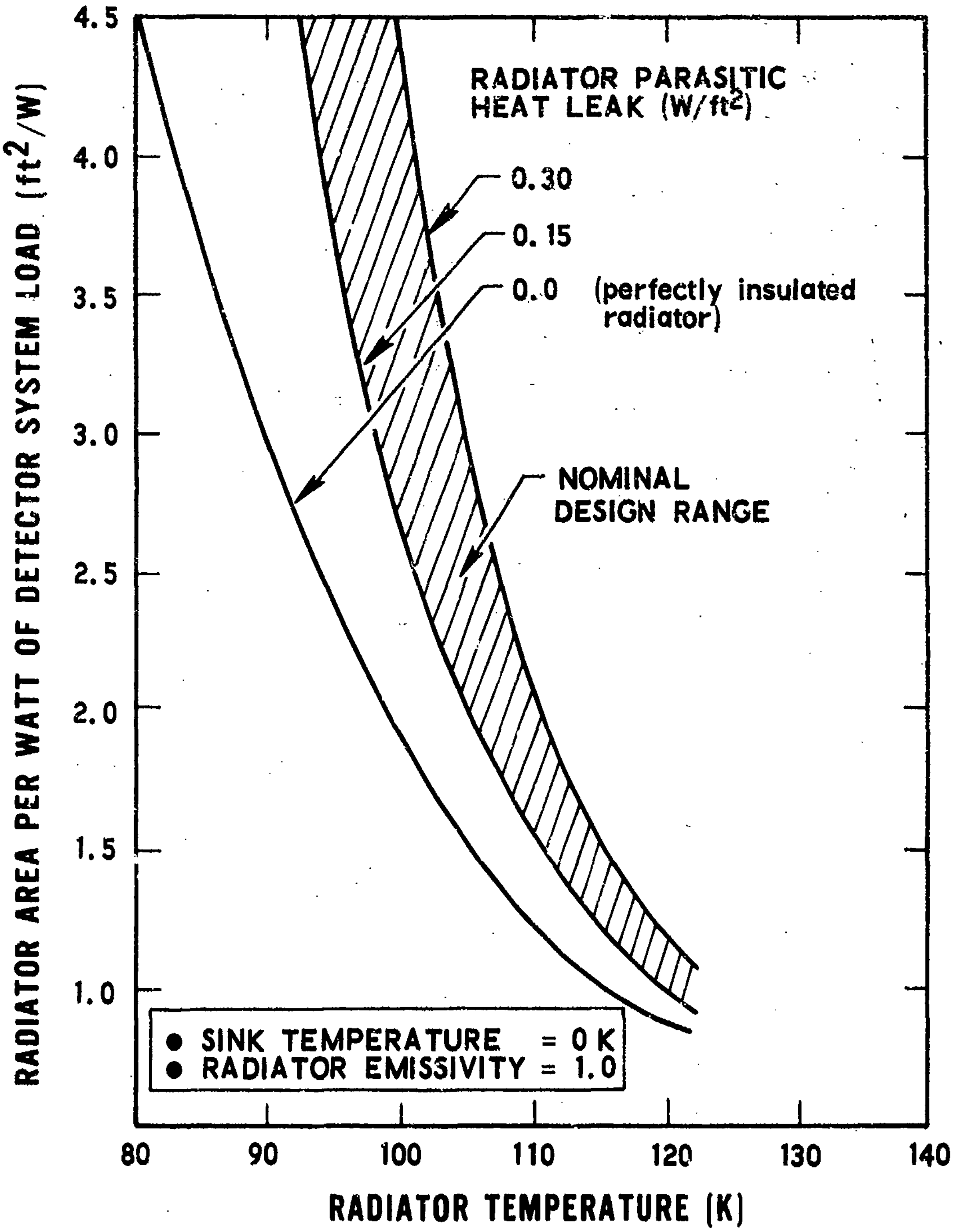


Fig. 4-2. Radiator Area per Watt of Detector System Heat Load Versus Temperature and Parasitic Heat Leak (Ref. 63)

Eight basic radiator designs (Figs. 4-3 and 4-4) encompassing single and multistage flat, parabolic and conical surfaces were analyzed (for further details see Ref. 64). Preliminary studies were made to select the best two or three configurations for each orbital case. From these selections more detailed analysis and considerations for development were evaluated. One radiator design was selected for each orbit condition. Further detailed analysis and designs for development testing were completed for a radiator applicable for use in an earth-oriented 200 nmi circular orbit.

Based on requirements and constraints established, the basic approach taken regarding the configuration and location of the radiators was as follows:

a. Temperatures of the radiators are passively controlled.

b. Shields are used to minimize heating of the radiator surfaces.

c. Location of the radiator on the spacecraft and the establishment of designs are dictated primarily by the particular orbit conditions and the orbital heating fluxes.

Preliminary analyses showed that the thermal performance of a staged radiator (i.e., more than one surface in series) is nearly maximized with three stages, and in some cases two stages. Additional stages provide negligible thermal improvement and add considerably to the complexity of the design.

The radiator designs are selected on their ability to attain low temperatures which is predicated primarily on the following factors:

a. The ability of the shields to minimize heating of the radiator by shadowing the radiator from direct solar, planetary, and spacecraft heating

b. The ability of the shields to redirect solar, earth albedo, and earth emission energy so that none or a minimum amount of this energy is incident on the radiator

c. The ability of the shields to attain as low a temperature as possible so that emitted energy from the shields to the radiator is minimized

d. The ability to maximize the view factor from the radiator to space

e. The ability of the multilayer insulation to minimize heat leaks from the spacecraft to the entire cooler

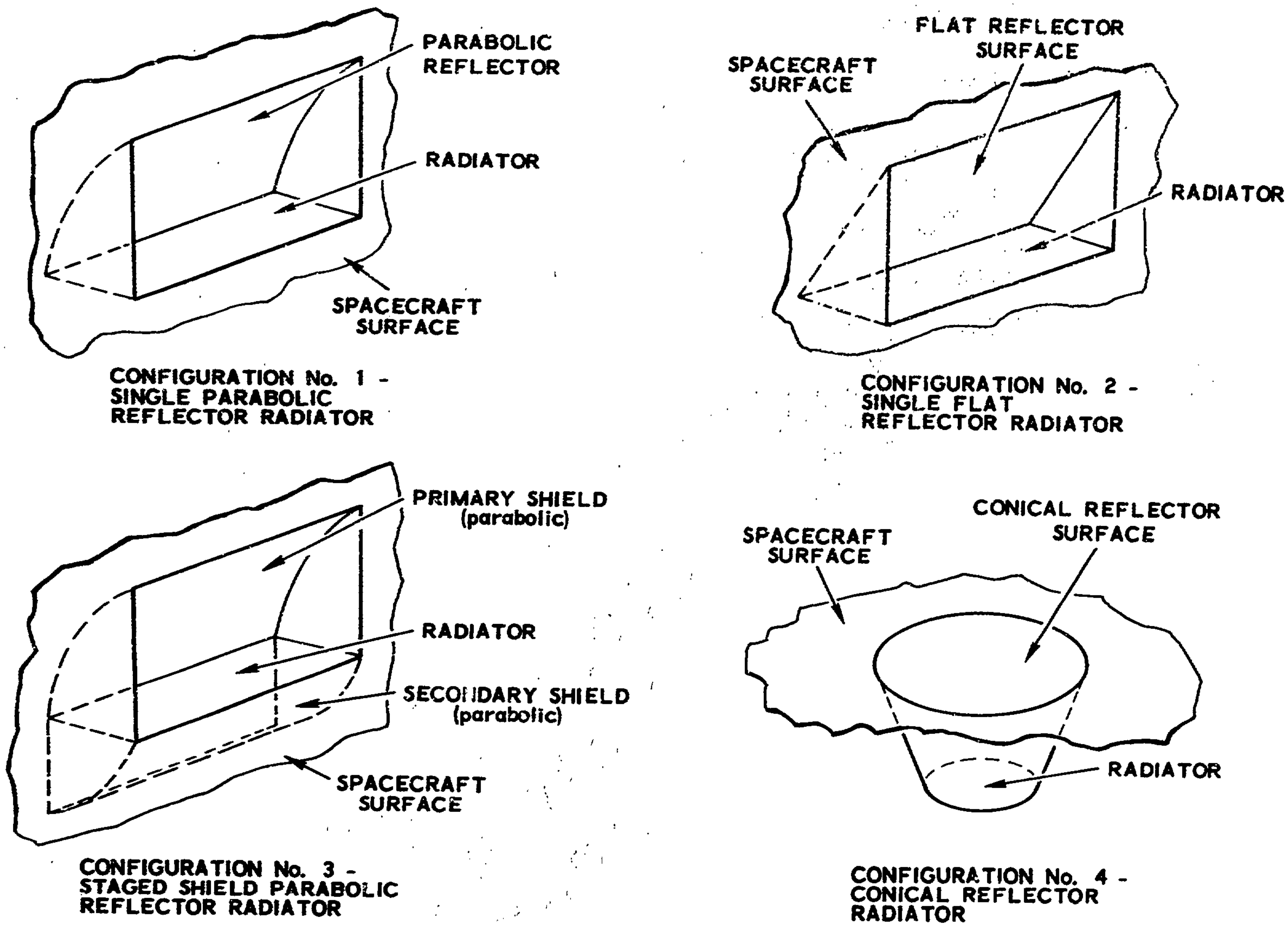


Fig. 4-3. Passive Radiance Cooler Designs for 77 K Operation (Configuration #1-4) (Ref. 64)

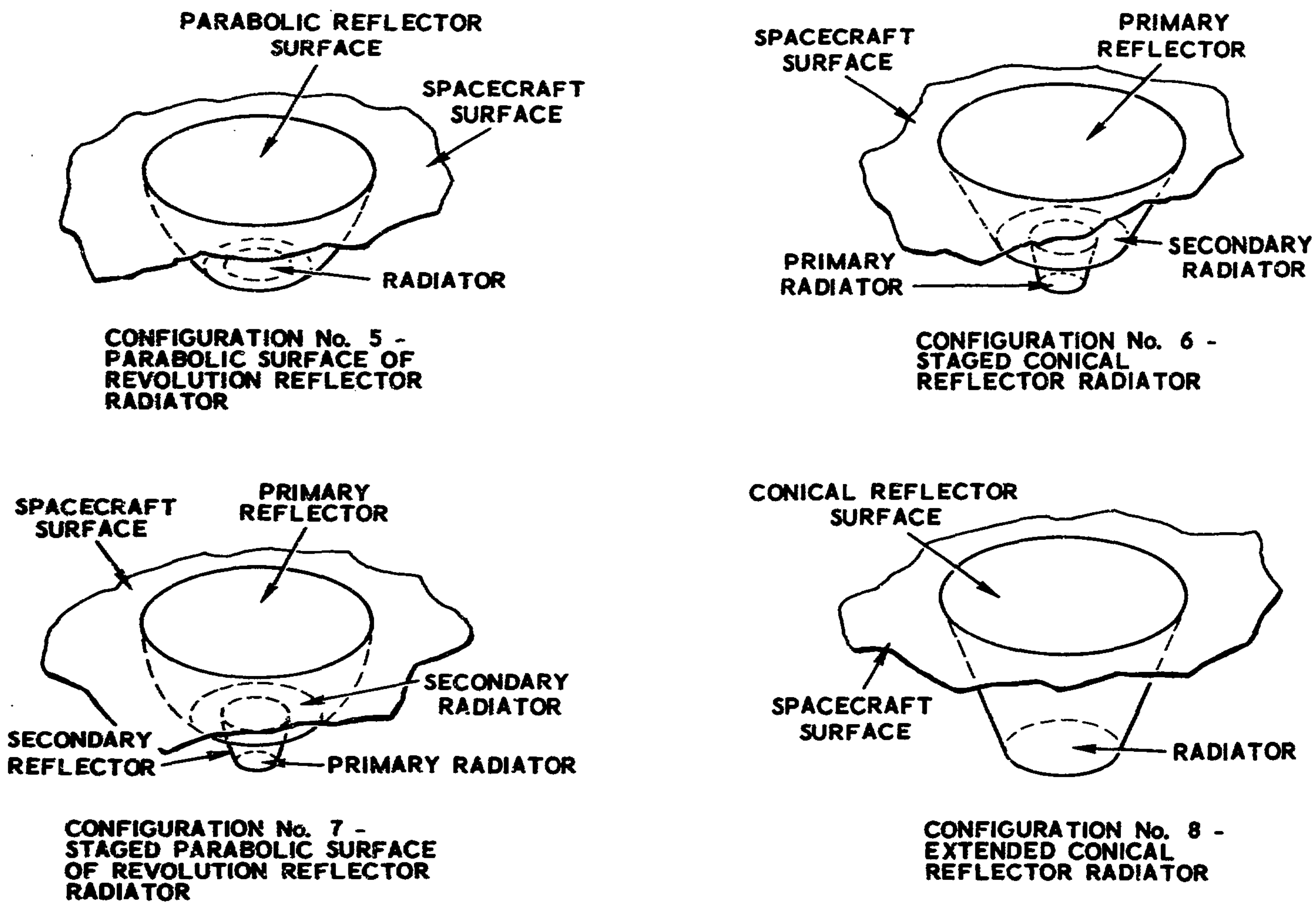


Fig. 4-4. Passive Radiance Cooler Designs for 77 K Operation (Configuration #5-8) (Ref. 64)

Based on the results of the AFFDL study, the achievement of a radiator temperature of 77 K or lower is strongly dependent on (1) the orbital parameters (including orbital altitude and orientation of the spacecraft with respect to the earth and sun), (2) shielding of the detector radiator from planetary and solar fluxes, (3) thermal isolation of the radiator from the spacecraft, and (4) the IR detector heat dissipation.

These studies have shown that the lowest radiator temperatures are attained with the following configurations for each of the three orbital conditions. The configuration selected for the 200-nmi circular orbit (Configuration #3, Fig. 4-3) is a staged shield, parabolic reflector-radiator. The radiator temperature for a 10 mW power dissipation is 85 K (153 R) for maximum planetary heating and 81 K (146 R) for minimum planetary heating. (Maximum heating is earth albedo and emission whereas minimum heating is earth emission only.) A unit conductance of 0.005 Btu/hr-ft^2-R is assumed for all configurations. Configuration #3 achieves a lower radiator temperature than either Configurations #1 or #2 because the radiator views a colder environment as a result of the double shields. For a given volume, however, Configuration #3 achieves a smaller radiator size. As a result, even though it is less sensitive to heating inputs from the earth due to the shields, it is more sensitive to heat inputs from the spacecraft. Because of the smaller area, its power dissipation would become a limiting factor before that in Configurations #1 or #2. In essence, these results show that the optimum radiator design is very sensitive to the spacecraft design criteria and to the tradeoff penalties associated with weight and volume and the required power dissipation and temperature.

The configuration selected for the 600-nmi orbit (Configuration #5, Fig. 4-4) is a parabolic surface of revolution reflector-radiator. The spacecraft is spinning in this case, and therefore a surface of revolution is required for the shields. As a result, the radiator useful volume is reduced from the 200-nmi orbit case; consequently, power dissipation densities increase.

The resultant radiator temperatures are therefore somewhat higher than in Configuration #3. The radiator temperature is 101 K (182 R) for maximum planetary heating and 96 K (172 R) for minimum planetary heating.

The configuration selected for the synchronous orbit (Configuration #6, Fig. 4-4) is a staged conical reflector-radiator. The restrictions are similar to the 600-nmi orbit. It has a range of radiation surface temperatures of 97 K (175 R) to 99 K (178 R) for minimum and maximum incident solar heating fluxes.

The development design configuration for the 200-nmi orbit is the staged shield parabolic reflector-radiator as shown in Fig. 4-3. The configuration has been fabricated and consists of a primary shield assembly, a secondary shield assembly, a staged radiator assembly, multilayer insulation assemblies, and support structure. An improved design has evolved which (1) meets the thermal requirements of the configuration; (2) maintains structural integrity at minimum weight; (3) incorporates materials and coatings which are compatible with the predicted space environment; (4) can be fabricated with state-of-the-art aerospace techniques; and (5) meets thermal vacuum chamber requirements for the testing of the configuration. The design is incorporated into drawings from which an experimental model for thermal vacuum (and structural) tests can be fabricated.

Based on detailed thermal analyses, the radiator for the development design configuration achieves a temperature of 81 K (46 R); the temperature of the radiator varies less than ± 0.3 R for one orbit. Based on recent personal communications (Ref. 65), experimental verification of the performance of the development design configuration in a simulated 200-nmi orbit will be accomplished with a thermal vacuum test using the Arnold Engineering Development Center, Tennessee, facilities later this year. The primary objectives of the thermal vacuum test are (1) to verify the ability of the configuration cavity to reflect incident earth albedo and emission energy and (2) to verify the thermal isolation of the configuration from the spacecraft.

2. NASA PROGRAMS

A number of scientific experiments in the NASA Application Technology Satellites (ATS) program which are to be placed in geosynchronous orbits will probably be using passive radiators. A number of satellites in this experiment series will have various IR sensor equipment aboard. The Optical Division of ITT, located in Fort Wayne, Indiana, has performed a number of studies for NASA (Refs. 66 and 67) on radiant coolers applicable to the Nimbus and Tiros programs. The analysis, design and test of a two-stage radiant cooler to operate near 77 K is reported in Ref. 67. This cooler was designed for mapping of the earth and its cloud cover from a near-polar orbit. A schematic of the cooler concept is shown in Fig. 4-5. The significant thermal loads on the first stage cone consist of earth IR, reflected sunlight and in some cases direct sunlight. The temperature of the first stage cone is controlled by a low α/ϵ ratio and is isolated from the spacecraft. The first stage patch (radiator surface) is thermally coupled to the first stage cone and to deep space. The second stage cone is designed so that the second stage patch views only the cone and deep space. The area of the second stage patch to which the IR detector is attached has an area of 3.8 in.^{2}. During simulated flight tests in a chamber, the second stage patch reached a temperature of approximately 80 K with a corresponding power dissipation of 5.65 mW. This work is continuing towards the development of passive radiators to cool IR detectors to temperatures in the 80 to 120 K regions for the NASA ATS program. A high resolution radiometer flown on Nimbus I, II, and III contains a single stage cooler that maintains a detector at 200 K. The cold patch has a radiating area of approximately 10 cm^2 and a radiative power of about 90 mW.

The Santa Barbara Research Center is currently working on two programs for NASA involving passive coolers (Ref. 36). An engineering model of a cooler designed to operate at 90 K for use in the Synchronous Meteorological Satellite has been tested on a preliminary basis and found to be within a few degrees of its design point. Another design under development is for a cooler to operate in the 90 to 100 K region in low earth orbit for use with the Earth Resources Technology Satellite.

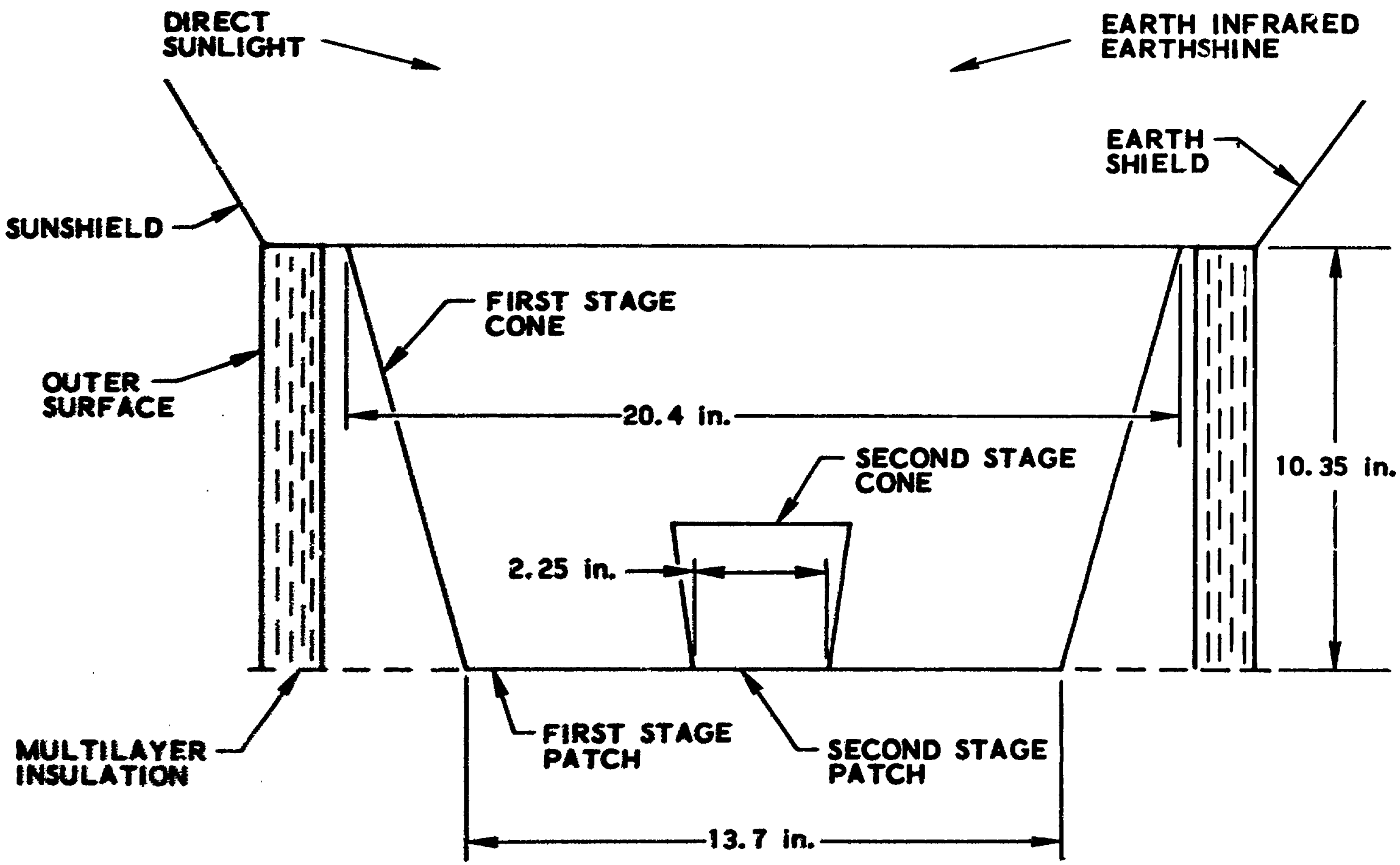


Fig. 4-5. Two Stage Rectangular 77 K Radiant Cooler (Designed by ITT for NASA) (Ref. 68)

V. THERMOELECTRIC COOLERS

A. BACKGROUND

When two dissimilar metals are connected in series with a source of emf, one junction will be cooled while the other junction becomes heated. The generation or absorption of heat depends on the direction of the direct current, while the rate of heat pumping is a function of the current and material properties. This basic phenomenon was first observed by Jean C. A. Peltier in approximately 1834.

This Peltier effect, which is distinct from the heating of a conductor by a current due to its resistance, is an inversion of an earlier observation made by Johann Seebeck. The "Seebeck effect" is the production of an emf in a circuit formed by different metals when their junctions are at different temperatures. This phenomenon, which is caused by the transfer of free electrons that are present in all metals is the basis of operation of thermocouples. Although the basic principles of this "thermoelectricity" are over 100 years old, development of practical heating or cooling devices utilizing these principles did not occur until the rapid progress in semiconductor materials occurred in the early 1950's. This progress in semiconductor materials, combined with the recognition by the U.S. Navy that these materials held the possibility of quiet power generation and cooling systems for ships, led to rapid advancement in materials up to the present time.

The outstanding features of thermoelectric coolers are simplicity and reliability, as there are no moving parts. Other advantages include:

a. The heating and cooling function can be easily interchanged by reversing the polarity of the direct current.

b. Noise is absent during operation.

c. Operation is independent of orientation or gravity.

d. Weight and volume are small.

The primary limitation of thermoelectric coolers is the maximum temperature difference (or temperature lift) attainable on a practical basis which is about 150 C based on available materials. The chief disadvantage of these devices is the relatively low coefficient of performance (COP) (i.e., cooling capacity divided by power input) where large temperature differences are required.

As a result, the prime areas of application are found where cooling loads are relatively low, space and weight available are small, large temperature differences are not required, extended periods of maintenance-free operation are necessary, and where power system weight penalties are small. Typical applications currently include cooling of IR detectors and various other electronic components, medical and laboratory instrument temperature control, and improvement of night vision sensitivity.

B. OPERATIONAL AND DESIGN CHARACTERISTICS

A typical single stage thermoelectric cooler consists of a p-type and an n-type semiconductor connected together by a metallic conductor as depicted in Fig. 5-1. (A p-type material has a shortage of electrons while an n-type material has an excess of electrons.) When a voltage is applied (normally in the range of 0.10 to 5 volts for typical IR cooling units), the flow of current in the direction shown produces a temperature difference between the two junctions by absorbing heat at one end and releasing it at the other. The heat removed from the cold junction is the difference between the Peltier cooling effect and the sum of (a) the Joule heat generated by the current, and (b) the heat conducted from the hot to the cold junction. The resultant is the net cooling capacity of the couple. If the two materials in the semiconductors are identical, the energy level of the electrons flowing would be the same throughout the system and there would be no heat pumping. When materials with different available electron energy levels are selected, the electrons flowing across the junction must undergo an energy change which results in either the absorption or rejection of heat.

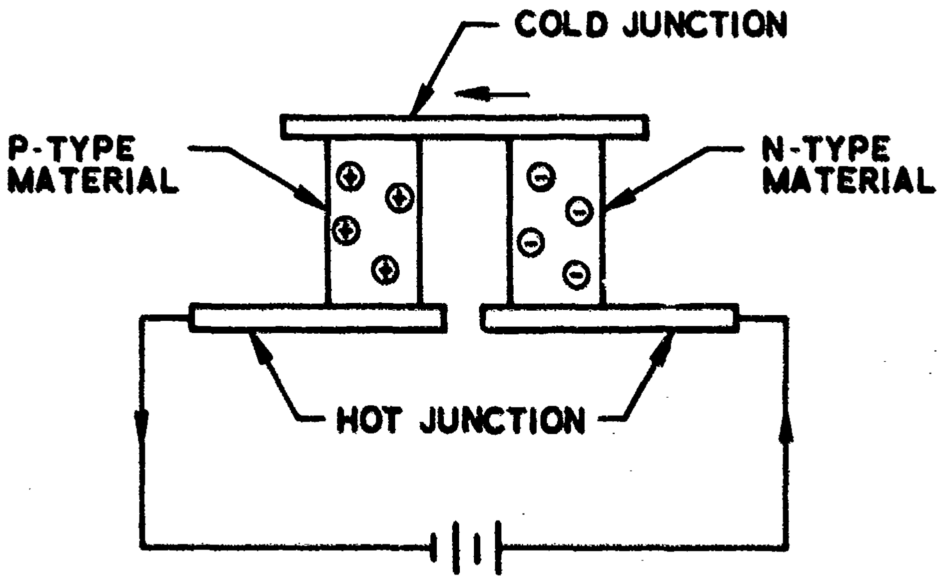


Fig. 5-1. Peltier Thermoelectric Couple

Thermoelectric couples of similar design can be arranged in parallel to increase the heat pumping capacity. When identical couples are placed in parallel and supplied with equal currents, they pump as many times more heat at the same temperature difference and with the same COP as there are couples. Thermoelectric couples can also be connected in series (cascaded) for one of two purposes: (1) to provide a temperature difference greater than that attainable from a single couple or (2) to achieve a higher COP for a given heat pumping rate and overall temperature difference. Manufactured units are generally available with up to four stages where temperature differences of 80 to 120 C are achieved between the cooling load temperature and the available heat sink temperature. Theoretical and actual performance data for various operating conditions are given in the following paragraphs.

C. THEORETICAL PERFORMANCE DATA

A brief review of theoretical performance parameters is presented here only to the extent necessary to assess the capabilities of thermoelectric coolers. For more detailed discussion of performance characteristics and derivation of the basic equations, the references cited in this report (Refs. 67 through 70) should be consulted.

When a steady state is established at the cold junction of a couple, the net heat absorbed at the cold junction less the Joule heat and the heat conducted from the hot end to the cold end is given by

$$Q_i = ST_c I - \frac{1}{2} I^2 R - K\Delta T \qquad (1)$$

where

Q_i = net heat absorbed at the cold junction (Peltier effect), W
S = Seebeck Coefficient for a couple, V/K
I = current, amp
R = electrical resistance, ohms
K = thermal conductance, W/K
T_c = temperature of cold junction, K

The first term on the right-hand side of Eq. (1) is derived from the fact that the Peltier cooling effect is given by

$$Q = P_{ab} I \tag{2}$$

and that

$$P_{ab} = ST_c \tag{3}$$

where P_{ab} = Peltier Coefficient between materials a and b (W/amp), and T_c = cold junction temperature (K). Thus

$$Q = ST_c I \tag{4}$$

All the equations given in the following paragraphs can be derived from the fundamental relationship given by Eq. (1).

The pertinent performance parameters of thermoelectric coolers which are most useful for evaluation purposes are:

a. The figure of merit (Z) which is a function of the material properties

b. The temperature difference or lift (ΔT) attainable between a single stage couple

c. The coefficient of performance (COP) which is equal to the cooling capacity divided by the power input.

The figure of merit (Z) serves to characterize a material for its cooling potential and is given for a single material by

$$Z = \frac{s^2}{\rho k} \tag{5}$$

where

s = Seebeck Coefficient for a single material, V/K

ρ = electrical resistivity, ohm-cm

k = thermal conductivity, W/cm-K

Z = figure of merit, K^{-1}

The Seebeck Coefficient (s) for a single material is the ratio of the emf produced per degree of temperature difference. Data on this ratio can be found in standard tables of Seebeck voltage series. The figure of merit for a couple can be expressed as

$$Z = \frac{(s_p - s_n)^2}{\left(\sqrt{\rho_n k_n} + \sqrt{\rho_p k_p}\right)^2} \tag{6}$$

The maximum temperature difference or lift (ΔT_{max}) between the hot and cold junction of a single stage couple is derived from Eq. (1) in Refs. 69 and 70 and is given as a function of the cold junction temperature (T_C) in degrees Kelvin as

$$\Delta T_{max} = \frac{1}{2} Z\, T_C^2 \tag{7}$$

An expression for ΔT_{max} as a function of the hot junction temperature (T_H) is given in Ref. 71 as

$$\Delta T_{max} = \left(\sqrt{1 + 2\, ZT_H} - 1\right)^2 / 2Z \tag{9}$$

This is sometimes more useful since in many appli ations the hot junction temperature is fixed. Using the above equations, one finds the maximum ΔT as a function of the hot junction temperature and Z is shown in Fig. 5-2a, while the hot junction temperature versus the cold junction temperature for various values of Z is shown in Fig. 5-2b. Although materials considerations are discussed in more detail later, the value of Z for the best pair of materials currently available on a production basis is in the range of 0.002 to 0.003, while materials in the development or experimental stage have Z values in the range of 0.004 to 0.005 at temperatures below 300 K.

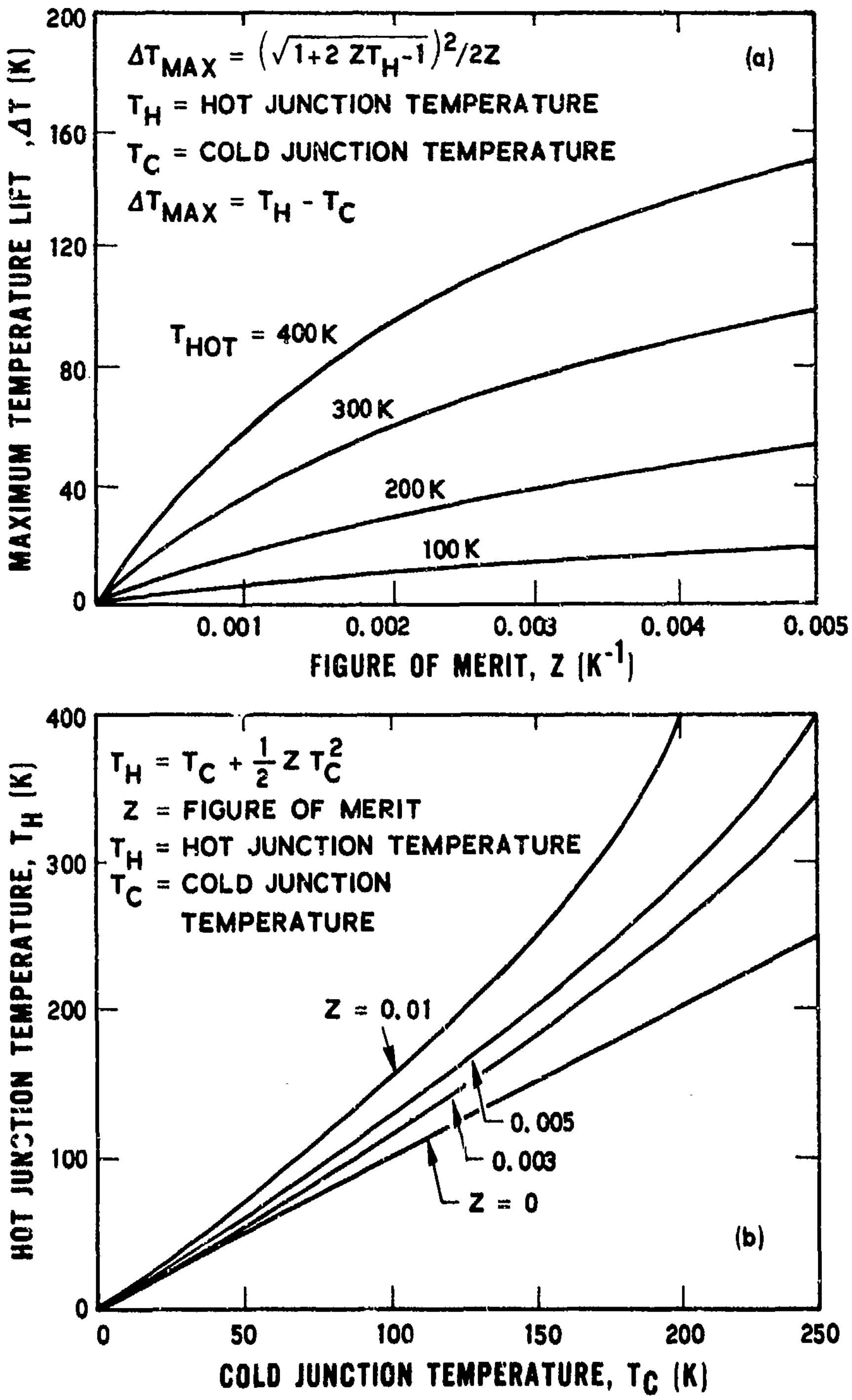


Fig. 5-2. Maximum Theoretical Temperature Lift and Minimum Cold Junction Temperature Attainable for a Single-Stage Thermoelectric Cooler

The COP is an extremely important parameter since the power supply is such a major consideration in space applications. In most cases it will be desirable to design the system to obtain the maximum COP (as opposed to designing for the maximum heat pumping rate). The maximum value of the COP of a single couple as a function of Z and the cold junction temperature (T_C) is given in Fig. 5-3 for a fixed hot junction temperature of 300 K. The COP is seen to decrease sharply with a decrease in T_C or a corresponding increase in ΔT.

The data presented thus far applies to a single stage couple. As previously indicated, by placing couples in series (cascading), a higher ΔT can be obtained. In this case, however, the heat rejected by each stage must be absorbed by the next one. Since the maximum temperature lift (ΔT_{max}) occurs only at a zero heat load, the actual attainable temperature lift for cascaded couples is only a fraction of the ΔT_{max} of each couple. The effect of cascading on the maximum COP is shown on Fig. 5-4. Figure 5-4a shows the variation in COP_{max} with the cold junction temperature for a fixed hot junction temperature of 300 K for one, two and three stages of cooling. Large gains from cascading are realized when the operating ΔT approaches the maximum ΔT attainable with a single stage.

The maximum ΔT attainable for a given value of Z by cascading is shown in Fig. 5-4b. It can be seen that each new stage greatly increases the maximum ΔT.

D. MATERIAL CONSIDERATIONS

The figure of merit, Z, which involves the three parameters previously defined (i.e., thermal conductivity, electrical resistivity and the Seebeck Coefficient), is the primary factor for evaluating materials for thermoelectric coolers. The properties which determine the Z value are interrelated and in general quite dependent on the electron density and the relative emf difference between two materials. The value of Z is normally maximized with electron densities associated with materials in the semiconductor class. These materials exhibit a high Seebeck Coefficient and relatively low values of thermal conductivity and electrical resistivity.

$$COP_{MAX} = \left(\frac{T_C}{T_H - T_C}\right) \frac{\sqrt{1 + Z\,\overline{T}} - T_H/T_C}{\sqrt{1 + Z\,\overline{T}} + 1}$$

T_H = HOT JUNCTION TEMPERATURE (K)

T_C = COLD JUNCTION TEMPERATURE (K)

Z = FIGURE OF MERIT (K^{-1})

$$\overline{T} = \frac{T_C + T_H}{2}$$

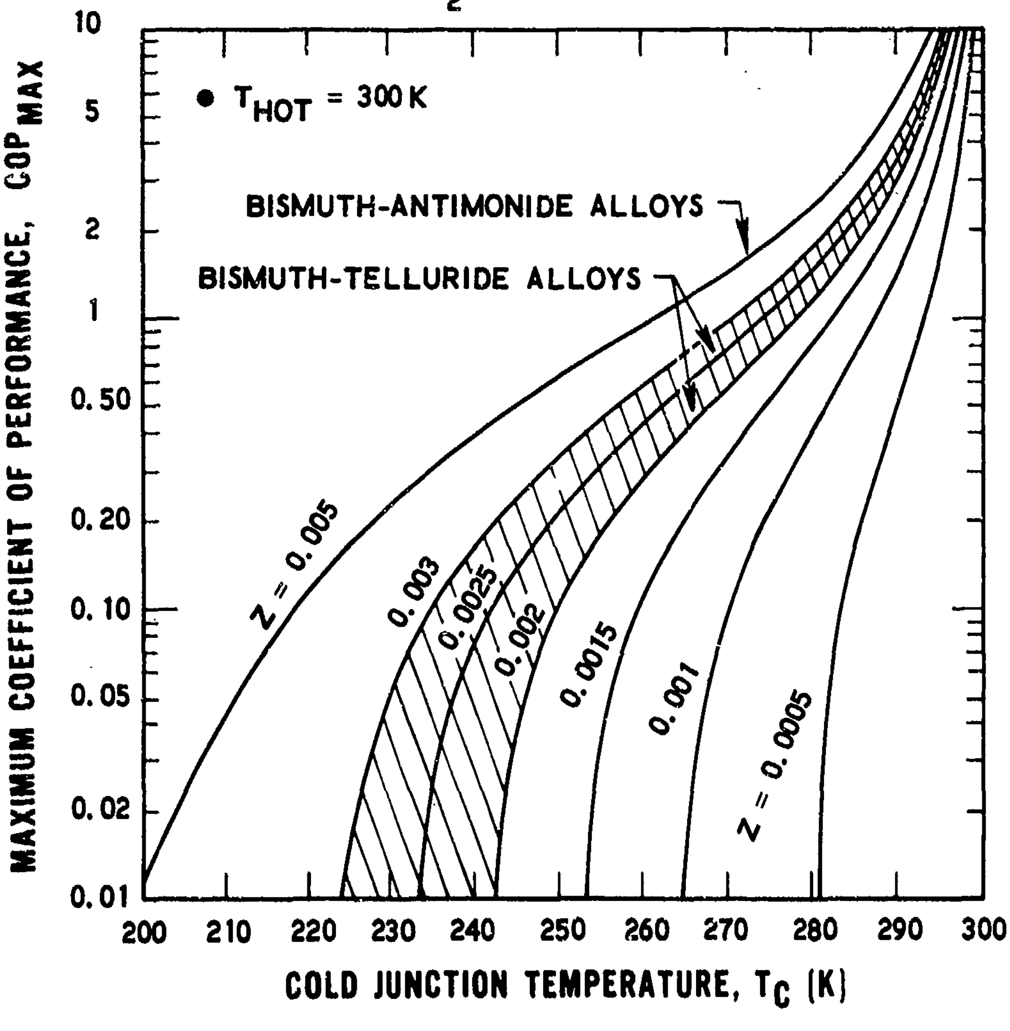
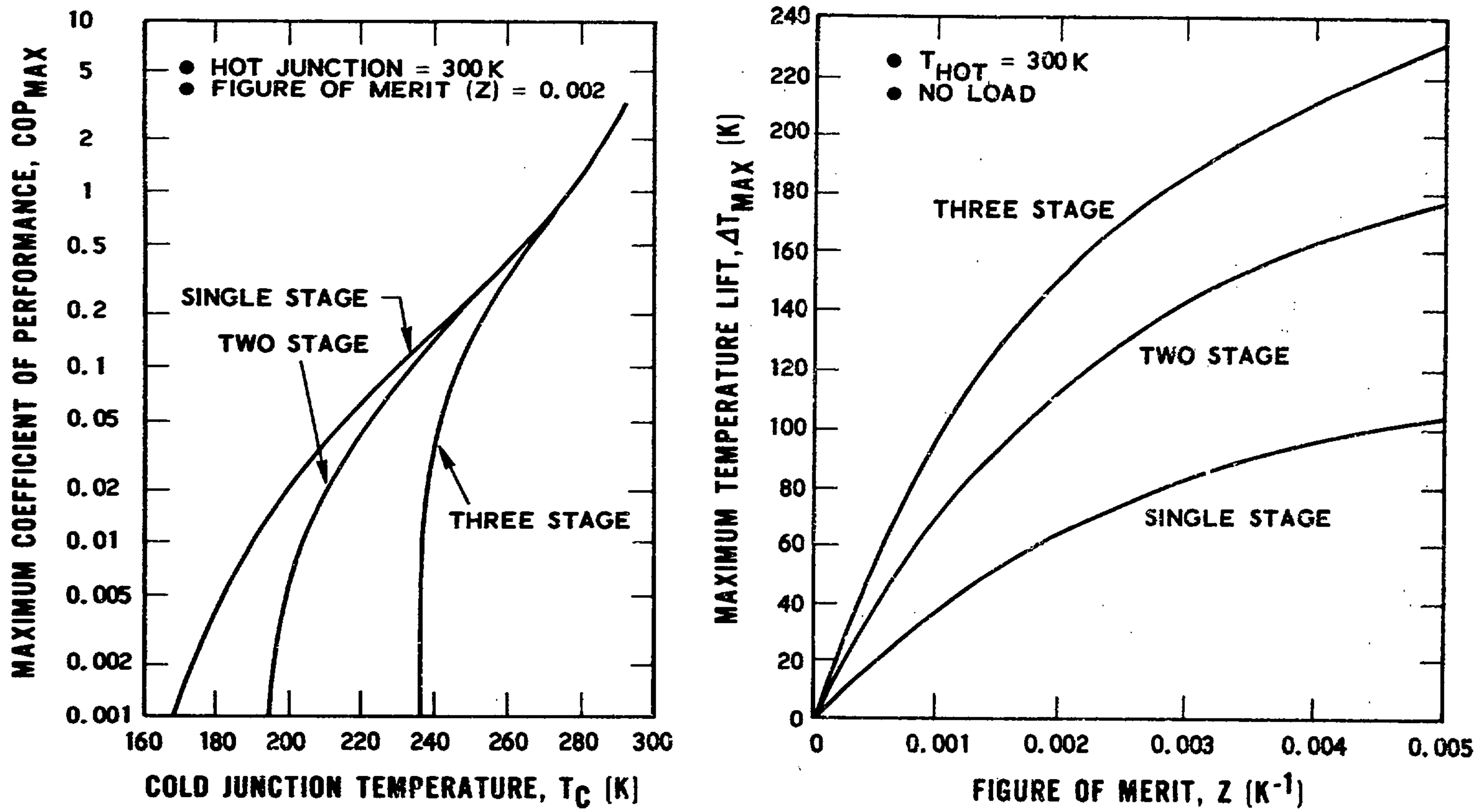


Fig. 5-3. Maximum Coefficient of Performance Versus Cold Junction Temperature and Figure of Merit for a Single Stage Thermoelectric Cooler (Ref. 71)



(a) MAXIMUM COEFFICIENT OF PERFORMANCE vs COLD JUNCTION TEMPERATURE

(b) MAXIMUM TEMPERATURE LIFT vs FIGURE OF MERIT

Fig. 5-4. Theoretical Performance of Multistage Thermoelectric Coolers

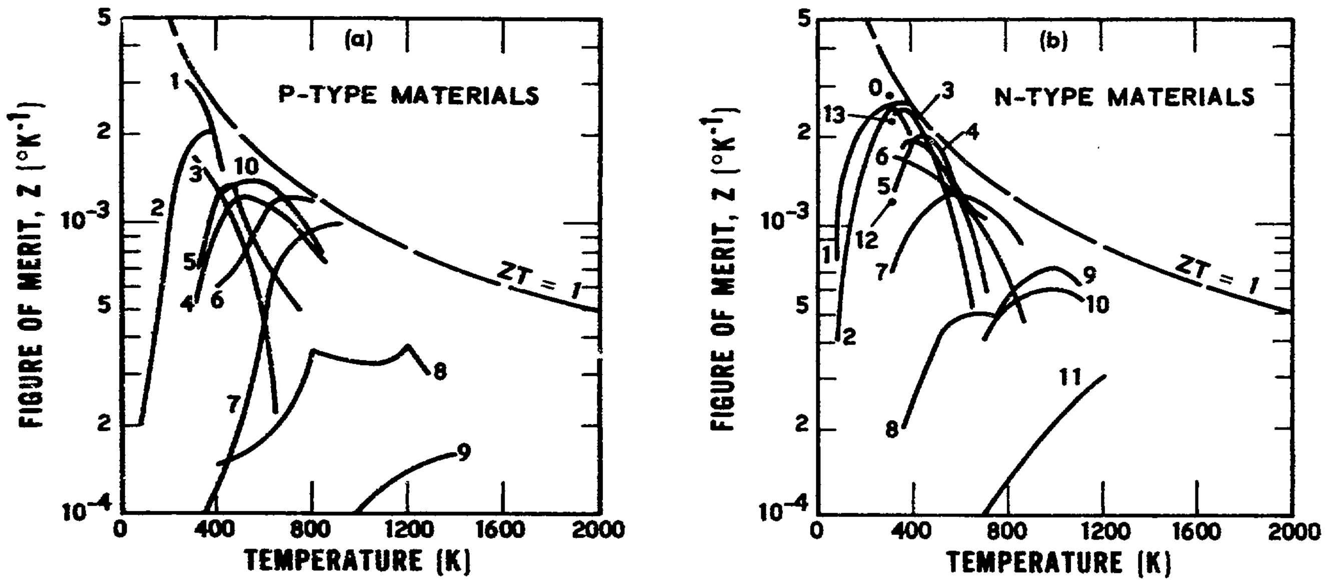
The limitation of thermoelectric cooling is primarily a materials performance limit. Data on the value of Z for various materials as a function of temperature is presented for both p-type and n-type materials in Fig. 5-5. In the region of interest for cryogenic cooling, relatively few materials are available.

At present, the best materials available and reproducible on a production basis are the Bismuth-Telluride (Bi_2-Te_3) alloys. Doping of pure Bi_2-Te_3 with various impurities provides the necessary positive or negative charge carriers to produce either p-type or n-type materials. The range of values for Z falls between 0.002 and 0.003 in the range of 200 to 300 K. Specific tabulated data on these alloys comparing theoretical versus actual test data are shown in Table 5-1. The ΔT_{max} is calculated using Eq. (9) based on measured properties with a hot junction temperature of 300 K.

In previous research and development programs most of the emphasis has been placed on a search for materials with high values of Z at higher temperatures which are suitable for power conversion systems. In the past few years, however, various other compounds have been developed with improved Z values at low temperatures, thus increasing the potential performance capabilities of thermoelectric coolers. Compounds of Bismuth and Antimony have been developed at the Lockheed Research Laboratories, Palo Alto, California (Ref. 72), under contract to AFFDL. The Z values in the range of 0.004 to 0.005 have been achieved at temperatures down to 200 K. This development is currently being funded by the U.S. Army Night Vision Laboratory, Ft. Belvoir, Virginia.

E. MANUFACTURERS AND DEVELOPMENT POTENTIAL

Companies most active in the manufacturing and development of thermoelectric coolers include Borg-Warner (Des Plaines, Illinois), Nuclear Systems (Garland, Texas), Cambridge Thermionic Corporation (Cambridge, Massachusetts), and Materials Electronics Products Corporation (Trenton, New Jersey) which is also known as MELCOR. A number of other companies



No.	MATERIAL
1	$BiSbTe_3$ + 0.56 W % Se
2	$BiSbTe_3$ + 0.20 W % Ag + 0.5 W % Se
3	PbTe + 0.1 at. % Na
4	PbTe + 1.0 at. % Na
5	ZnSb
6	$Ge_{0.9}Bi_{0.1}Te$
7	GeTe
8	$Na_{0.01}Mn_{0.99}Te$
9	$Li_{0.05}Ni_{0.95}O$
10	PbTe

No.	MATERIAL
0	75-25Bi_2Te_3 - Bi_2Se_3 + 0.05 W % Cu Br
1	80-20 Bi_2Te_3 - Bi_2Se_3 + 0.05 W % Cu Br
2	Bi_2Te_3 + 0.085 W % Cu Br
3	$Bi_2Te_{2.4}Se_{0.6}$ + 0.05 W % Cu Br
4	Bi_2Te_2Se + 0.05 W % Cu Br
5	PbTe + 0.5 W % EXCESS Pb + 0.2 W % Bi
6	PbTe + 0.02 mol % PbI_2
7	PbTe + 0.10 mol % PbI_2
8	InSb
9	In $As_{0.9}P_{0.1}$
10	In As
11	$CeS_{1.37}$
12	PbS
13	Ag_2Se

Fig. 5-5. Figure of Merit for Thermoelectric Materials Versus Temperature

Table 5-1. Thermoelectric Data for Various Bismuth Telluride Alloys (Ref. 71)

Material	Impurity Added	Conductivity Type	Figure of Merit (Z), x 10^{-3}	ΔT_{max} (C)	ΔT_{meas} (C)
Bi_2Te_3	CuI	N	2.5	65	59
Bi_2Te_3	Bi	P	2.2		
Bi_2Te_3	AgI	N	2.6	66	60
Bi_2Te_3 - 25% Sb_2Te_3	Bi	P	2.2		
Bi_2Te_3 - 25% Bi_2Se_3	CuBr	N	2.7	67	62
Bi_2Te_3 - 25% Sb_2Te_3	Bi	P	2.2		
Bi_2Te_3	AgI	N	2.6	68	65
Bi_2Te_3 - 20% Sb_2Te_3 - 5% Sb_2Se_3	Bi	P	2.4		

active at one time but not now involved in this area include RCA and the Westinghouse Electric Corporation. Characteristics of typical production units designed for cooling of IR detectors or other electronic components are presented in Table 5-2. This is not an all inclusive list, but represents a cross-section of typical units available and their corresponding performance capabilities.

All of the units shown in Table 5-2 utilize the basic Bismuth-Telluride alloys with various doping compounds. The system design limitations are shown in terms of the maximum ΔT attainable (corresponds to a no-load condition), the corresponding cold junction temperature for a 300 K (+27 C) hot junction, and the maximum cooling load possible (as ΔT approaches 0.0). System performance characteristics are shown for a nominal cooling load at a specified cold junction temperature, the corresponding ΔT, power input required, and resulting COP. The relationship between the cold junction temperature, heat load, and power input for these units is typified by Fig. 5-6.

All the units shown are standard production items with the exception of the two Nuclear Systems units noted (Models 2DG and LT700). These units were developed for the Army Night Vision Laboratory and are currently undergoing testing in prototype Army Night Vision systems. The Army Night Vision Laboratory (Ft. Belvoir) is currently funding Nuclear Systems and Borg-Warner for development of thermoelectric coolers for use at approximately 145 K in a terrestrial environment. Experimental units built to date (an eight-stage cascaded system) have operated at 145 K (under no-load conditions except for the environmental heat input) with a hot junction temperature of 300 K. Input power required is approximately 50 W. Plans for the next phase of the program include performing essentially under the same conditions with the addition of a 100 mW heat load (Ref. 76).

Table 5-2. Characteristics of Typical Production Thermoelectric Coolers

Manufacturer/Developer and Model Number	System Design Limitations[a]			System Performance Characteristics[a]				
	ΔT_{max} at No Load (C or K)	Cold Junction Temp. at ΔT_{max} (K)	Maximum Cooling Load (Q_{max}) at $\Delta T = 0.0$	Nominal Cooling Load	Power Input (W)	ΔT (C or K)	Coefficient of Performance (COP)	Number of Stages
Borg-Warner (Ref. 73)								
837	70	230	1.2 W	150 mW at 240 K	1.75	60	0.085	1
920	70	230	19.	5.5 W at 250 K	27.2	50	0.202	1
950	66	234	30.	10 W at 255 K	50.	45	0.20	1
970	70	230	130. W	30 W at 252 K	175.	48	0.172	1
447	90	210	200. mW	15 mW at 215 K	3.8	85	0.004	2
638	90	210	800.	200 mW at 220 K	6.0		0.033	2
623	117	183	140.	20 mW at 200 K	7.5	100	0.0027	3
724	100	200	900.	------	-----	----	------	3
493	95	205	90.	10 mW at 230 K	1.1	70	0.009	3
670	119	181	200. mW	10 mW at 185 K	5.4	115	0.0019	4
605	126	174	------	10 mW at 178 K	6.0	122	0.0016	4
Cambridge Thermionic Corporation (Ref. 74)								
800-3955	77	223	------	1 W at 240 K	29.4	60	0.034	2
800-3956	84	216	------	100 mW at 220 K	28.0	80	0.0036	3
800-1004	90	210	4.3 W	1.0 W at 236 K	29.6	70	0.034	2
800-1005	85	215	5.7 W	1.0 W at 228 K	24.5	72	0.041	2
800-1006	105	195	1.6 W	100 mW at 200 K	13.5	100	0.0074	3
Nuclear Systems Inc. (Ref. 75)								
SHF2	87	213	------	1.0 W at 243 K	6.5	57	0.154	2
SHF28	112	188	------	260 mW at 203 K	4.4	87	0.04[illegible]	4
ZDC89087	115	185	------	40 mW at 193 K	5.5	107	0.0073	4
4FH0259	117	183	------	100 mW at 195 K	7.0	105	0.0123	4
ZDC[b]	107	193	------	------	6.0	---	------	4
LT700[b]	130	170	------	------	24.0	---	------	6

[a]Based on hot junction temperature of 300 K (+27 C) and a vacuum of 10^{-4} to 10^{-6} mmHg

[b]Prototype models developed for U.S. Army Night Vision Laboratory, Virginia (Ref. 76)

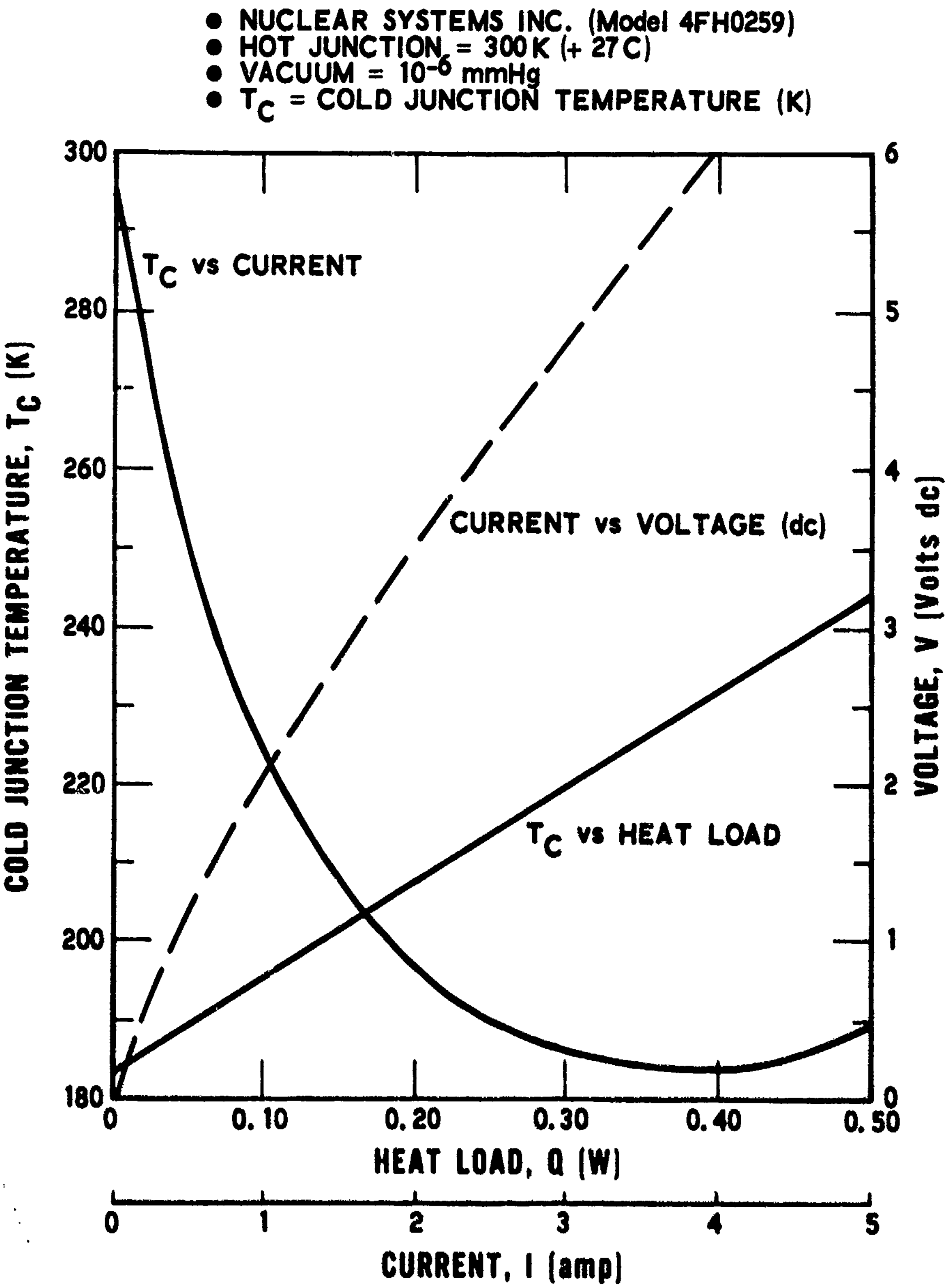


Fig. 5-6. Typical Operating Characteristics of Thermoelectric Coolers

Experimental units developed by RCA for the AFFDL in 1968 produced 10 mW of loading at approximately 140 K with a power input of 200 W. Subsequent units developed by RCA to operate with power inputs of 50 W or less produced 50 mW cooling at a minimum temperature of 180 K. This effort was discontinued.

The chief limitation of thermoelectric coolers for space applications is the figure of merit attainable. As a result, the high power requirements and limited temperatures available make thermoelectric coolers competitive with other techniques only at very small heat loads. With the utilization of technology and materials being developed under current Army programs coupled with the use of passive radiator systems to reduce the hot junction temperature, operation of thermoelectric coolers down to near 100 K may be feasible in the near future.

VI. CONCLUSIONS

A. CLOSED-CYCLE MECHANICAL REFRIGERATORS

Background data, description and operation of various refrigeration cycles, performance data and development potential have been summarized for a number of refrigeration systems applicable to spaceborne operations.

Stirling cycle refrigerators possess several of the primary requirements for spaceborne refrigeration systems (i.e., low power consumption and small size and weight). Although present designs have a limited life (500 to 1000 hr) due to wearing of seals and bearings, development of a Stirling refrigerator capable of operating continuously for 20,000 hr or more in a spacecraft appears to be feasible. A modified commercial unit has been operated successfully in space.

The VM cycle refrigerator, currently under development for use in spacecraft, is considered to have better life potential than the Stirling cycle refrigerator because of low internal loads on bearings and seals but requires more power. The potential for supplying energy directly in the form of heat using radioisotope or solar energy is another advantage for spacecraft applications. A development unit was recently operated successfully in space.

The Gifford-McMahon/Solvay cycle refrigerator utilizing separated components has been well developed because of the commercial attractiveness of this approach. The primary limitation of these systems for spaceborne applications is that substantially more power is required than for the Stirling or VM systems, and system weights are significantly higher primarily due to compressor requirements. Nevertheless, these systems provide the longest maintenance-free operating lifetimes currently available (i.e., up to 3000 hr).

Development is in progress to design gas-bearing turbomachinery utilizing the reversed Brayton and Claude cycles. Turbomachinery units appear to have the best potential for long operating life as well as a minimal

development risk. Although there is little hardware data available, the high power requirements (due primarily to the low efficiencies of the turbo compressors and expanders) make this refrigerator competitive with other refrigerators only at low temperatures (below about 20 K) or at higher capacities (20 to 100 W) where efficiencies improve somewhat relative to the other types of refrigerators. It is not likely that complete turbomachinery refrigeration systems applicable to military or space systems will be available for several years.

Work is also in progress to develop small rotary-reciprocating machinery utilizing the reversed Brayton cycle. In theory, this concept has the most promise for long-life operation and potentially can result in the minimum power and weight at temperatures below about 20 K. The complexity and novel approach of the rotary-reciprocating refrigerator represents a development risk, however, and only one unit has been built to date.

The closed-cycle J-T system has been extensively developed for specialized aircraft applications, but not for spacecraft useage because of its high power requirements.

B. OPEN-CYCLE EXPENDABLE SYSTEMS

The open-cycle J-T system provides the simplest, least expensive approach to short-term (usually measured in hours) cryogenic cooling. The primary drawback is the weight penalty for storage of high-pressure gas where extended operation is required. Data on typical units and theoretical and production hardware data are presented for estimation of gas storage penalties.

Cryogenic fluids stored as liquids in equilibrium with their vapors (subcritical) can provide a convenient constant temperature control system for ground-based or advanced aircraft and spacecraft applications in the range of approximately 4 to 90 K. Data for selected cryogenic liquids and various system design characteristics including storage penalties are summarized.

Cryogenic fluids stored at pressures above their critical pressures (supercritical) as homogeneous fluids can provide cooling at various temperatures for use in space operations. Supercritical helium, for example, used in this fashion for cooling in the range of 10 to 100 K is competitive with spaceborne closed-cycle refrigeration systems for up to approximately 60 days continuous operation. Cryogenic tankage under development could extend this time to six months to a year or longer.

Solidified cryogenic coolants can provide a reliable, lightweight, low-power system for small wattage heat loads, in the range of 10 to 125 K, for a year or longer. Laboratory models have demonstrated the feasibility of such systems; however, limitations involve restrictions on detector mounting and requirements for specialized filling procedures.

A limited number of fluids exist which can be stored at room temperature and manipulated thermodynamically to provide cryogenic cooling. A system utilizing ethane stored at 80 F and 631 psia would be capable of providing cooling at 100 K for approximately 0.10 lb/W-hr.

C. PASSIVE RADIANT COOLERS

Cryogenic radiators utilizing the low temperature sink of space directly produce an attractive, completely passive cooling system capable of high reliability for extended periods. Descriptions of several radiators designed to provide cooling in the range of 70 to 100 K with capacities up to approximately 10 mW are presented.

D. THERMOELECTRIC COOLERS

Thermoelectric coolers utilizing the Peltier effect prov[illegible] a simple, lightweight, reliable method of cooling fractional wattage loads in the region of 200 K. Production [illegible] are available in capacities ranging from 10 mW at 180 K [illegible] (both based on a heat sink of 300 K).

The primary limitations of thermoelectric coolers (due to materials properties) for space application are the high power requirements and the limited temperatures achievable. Multistaged coolers are under development for operation at 145 K in a terrestrial environment. If utilization of this technology is coupled with low temperature space radiators, operation of thermoelectric coolers down to near 125 K may have practical space application in the near future.

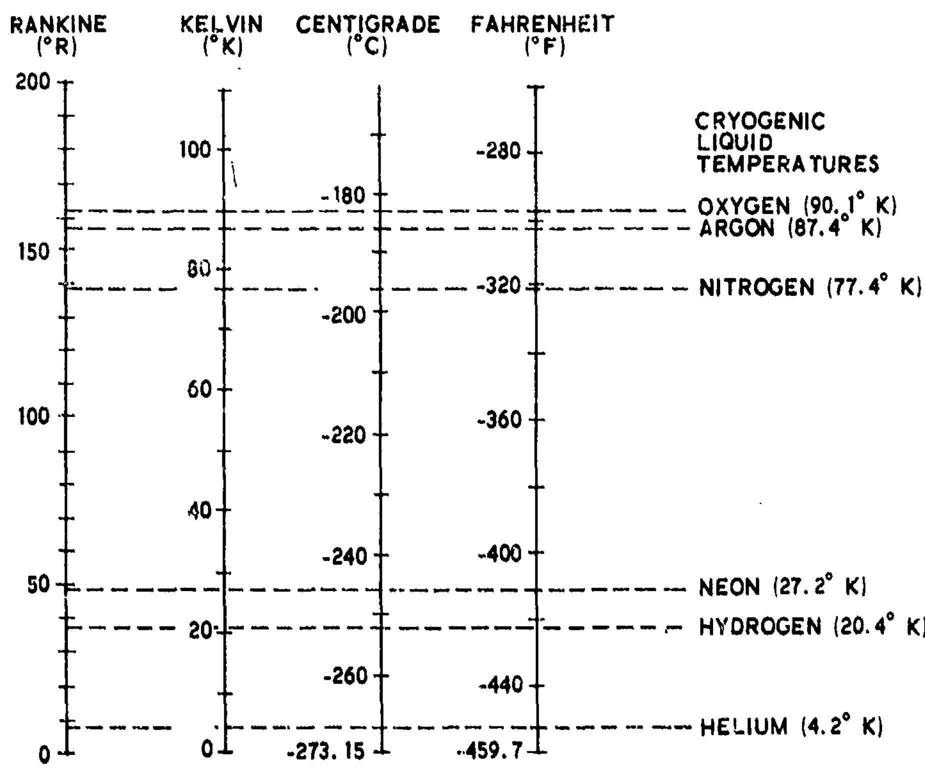
APPENDIX

TEMPERATURE SCALES AND CONVERSION

Temperature Conversion

To convert from units below to those on the right, perform the indicated operations in order	C	F	K	R
C	x 1	x $\frac{9}{5}$ + 32	+ 273	x $\frac{9}{5}$ + 492
F	- 32 x $\frac{5}{9}$	x 1	x $\frac{5}{9}$ + 255	+ 460
K	- 273	x $\frac{9}{5}$ - 460	x 1	x $\frac{9}{5}$
R	x $\frac{5}{9}$ - 273	- 460	x $\frac{5}{9}$	x 1

Temperature Scales



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